MACHINERY.

Vol. 5. April, 1899. No. 8.

THE MACHINERY OF A GREAT OFFICE BUILDING.

THE ELEVATORS, ENGINES, GENERATORS AND HEATING SYSTEM OF THE HIGHEST OFFICE STRUCTURE IN THE WORLD.

In our modern commercial centers the immense values attached to ground space demand that buildings erected for business purposes, must, to be profitable, contain a maximum amount of space for a minimum of ground area. The only apparent solution of this problem seems to be upwards—in other words, to build these structures to such heights as to fill the conditions required to make paying investments of the ventures.

That this method has been followed in New York is self-evident, as one of the first and strongest impressions of a visitor to the metropolis is produced by the number and immensity of the modern "sky scrapers," which have become such a prominent feature of the lower part of Manhattan Island. There are at present nearly two hundred office buildings in this city, besides many others, which, originally intended for manufacturing purposes, have been partially given over to office use.

From an architectural standpoint, these huge piles of stone, brick and iron are particularly exposed to criticism, and by many able critics are condemned as lacking in beauty of design and as having the appearance of one story piled upon another, without regard to any of those peculiarly pleasing and artistic effects so often seen in smaller buildings. The criticisms undoubtedly contain some truths, but it must be remembered that these structures are designed strictly for utilitarian purposes, and where questions of art and usefulness conflict the architect must obey the will of the engineer.

Since these vast creations may be considered as belonging to the realm of engineering, a description of some of the appliances in them will probably be of interest to the readers of this paper, especially as the peculiar conditions have called for many novel schemes and ingenious methods of applying the principles of mechanics.

The subject of this description will be one of the most notable of the late additions to the large office structures—the Ivins Syndicate Building, on Park Row, a half-tone of which is shown on our first page. By way of introduction, it may be stated that this building cost about $2,400,000, and contains nearly 8,000 tons of structural steel, which forms the skeleton or framework. Held in place and supported by this steel frame are 12,000 tons, more or less, of stone, brick and terra cotta, which make up the skin or covering, and give some scope for architectural and artistic effect. Owing to the high prices of land and the fact that some of the adjacent property owners asked prohibitive prices for their holdings, the plan of the structure is not as originally intended. It is now of quite complicated shape, and has much larger proportions than the front view would indicate. It has a frontage on Park Row of nearly 104 feet, but its extreme depth is nearly double this measurement. The height of the main building, exclusive of the towers, is 309 feet, and this is the rise required of the elevators running to the twenty-sixth floor.

Photo. by Reichert & Henius, 136 Liberty St., N. Y.

Ivins Syndicate Building, New York.

A habitation at a height of three hundred feet from the earth has some remarkable and many agreeable features. In summer the conspicuous absence of the noise, dust and flies that are so annoying at lower levels is a desirable feature, while the view in this particular instance is probably unparalleled in New York. It would, however, be a breath-taking climb to reach these lofty eyries were it not for the modern elevator system, and there can be no doubt that without this efficient service the present height of these buildings would never have been reached.

Every department of engineering has its own peculiar features and difficulties to overcome, and the elevator problem presents not a few. One of the first requirements is safety, and then follow speed, cost of maintenance, ease of regulation, comfort, etc. These features are met more or less successfully by the various companies building elevators for passenger service, but probably

the instalment of the Sprague Electric Elevator Co. in this building will compare favorably with any other in the country. The conditions imposed in this particular instance of speed, great rise and limited space for the elevator mechanism, are probably not equaled, and could scarcely have been as successfully met by any other than their special style of machine.

Through the courtesy of the representatives of the Sprague Company we are enabled to present the accompanying cuts, which, with the description, will make the leading features of this installation plain. Figs. 3, 5 and 6 show their vertical-screw machine, which has the electric motor directly connected to the vertical screw, as shown in Fig. 3, and which thus makes as compact a mechanism as can be imagined. Working on this revolving screw is the stationary ball-bearing nut, shown just above the hoisting machine, in Fig. 3, and detailed in Fig. 2. This mechanism is a remarkable achievement, and one that a designer or machinist would probably pronounce as totally impracticable if he had never seen or heard of its practical success. There can be no doubt, however, that it would be impracticable if deprived of the high-grade tools necessary for its production, and which, by the way, are liberally provided in their finely equipped shop at Bloomfield, N. J.

the saddle is made conical, to fit the shape of the nut at E, Fig 2, and the friction between the bronze facing of the nut at E and the conical recess in the saddle is sufficient to prevent the nut from turning while carrying the load under ordinary circumstances. When the end of the travel is reached, however, should the operator neglect to stop the motor, the nut and screw will turn together without harm, until the motor is stopped.

Fig. 9 shows the combination india-rubber buffer stop, and its position in the system is indicated in Fig. 3, just below the traveling multiplying sheaves. This device is designed to absorb the shock of the elevator car when descending, should it overrun the stopping place at the ground floor; but if the automatic stop be accurately adjusted, the buffer comes into use only in emergencies.

The travel of the car and the ball-bearing nut is in the proportion of 16 to 1, but we are assured that this bears no relation to the much-discussed bi-metallic standard, being merely the best proportions for this system. This ratio is obtained through the fixed and traveling multiplying sheaves, shown in Fig. 3, which are provided with roller bearings, as shown in Fig. 7. The use of ball and roller bearings is continued throughout the machine wherever practicable. In Fig. 8 is seen the roller retain-

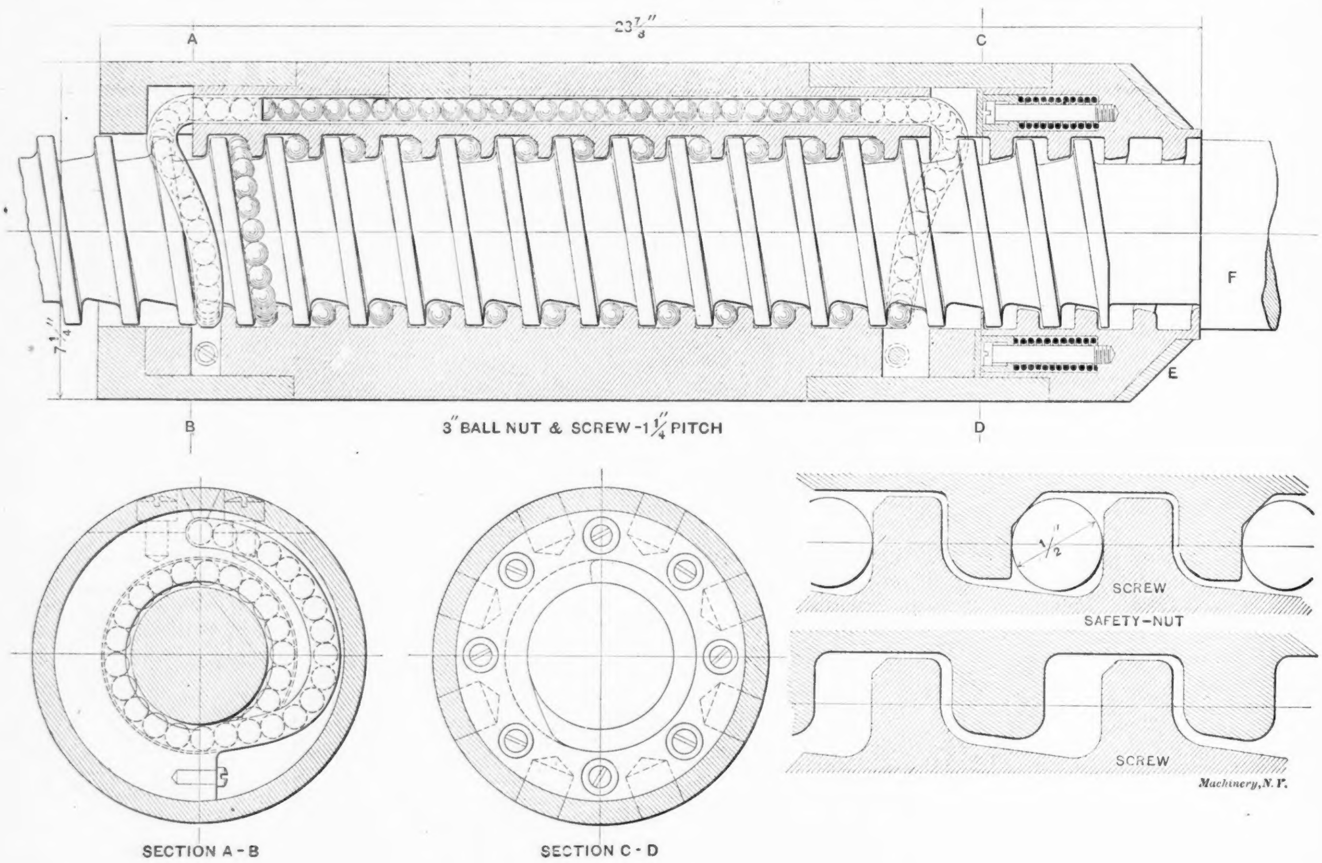


FIG. 2.

It will be readily seen that a variation of the pitch of thread, either in the nut or screw, would throw an excessive load on a few balls, and probably result in cracking them and throwing the mechanism out of order. The body of the nut is made of Harveyized steel, and is ground in the threads by a special internal grinding rig that corrects the inaccuracies resulting from the hardening process, and leaves the thread uniform in pitch and angle. The screw, which is turned from high carbon steel, is left unhardened, as no satisfactory process is known for successfully handling pieces of such length, and besides, with the proportions given by the designer, the wear of the thread proceeds very slowly. In Fig. 2 the shape of the thread is shown, nearly full size, with a ball in position. The ball-bearing nuts used in this installation contain about 380 one-half inch balls, and as less than 80 are required for the return route (which is clearly shown in the figure), there are constantly over 300 balls carrying the load.

In plan B, Fig 4, the saddle is shown that carries the nut and transmits the motion to the multiplying sheaves by means of the rods shown in section on each side of the center. The recess in

ing disc for the rollers which take the thrust of the screw, and is again shown in section at Figs. 5 and 6, on a line with the junction of the lower cap and the main portion of the motor. This cap is made oil-tight, so that the thrust bearing is always flooded with oil.

It will be observed that the screw is always in tension when carrying the load, so that no trouble is ever encountered by buckling. This feature permits the use of a screw of as small diameter as is consistent with tensile strength. So easily does this mechanism operate that in lowering the load, it runs down by the force of gravity without necessitating the reversal of the motor.

It is only when raising the load that the motor uses the electric current as in descending the revolving screw and connected armature turn backward freely. To prevent the car from descending too rapidly the field-magnets are excited from the line wire and the armature terminals are short-circuited through a variable resistance under the control of the elevator attendant. He thus has an electric brake at his command which has a range from a gentle check on the descent, to a complete re-

versal by simply varying the resistance. A brake shown in Figs. 5 and 6 is provided, but is only used for holding the car in any desired position, as at a floor level, the checking of the motion being entirely effected by varying the resistance. The result is a smooth and agreeable motion, even at the high speed at which the cars travel in the Ivins Building, which is given as 600 feet per minute. There being the absence of the disagreeable jolts so frequently experienced in the elevator service of some buildings at much lower speeds. At the quoted rate of travel the vertical distance from the street level to the twenty-sixth floor can be covered in about one-half minute if no stops be made on the way.

A heavy and constantly varying load is carried by these motors as will be readily understood when it is remembered that they are required to start the car at a ratio of 16 to 1 and get it quickly into motion at the rate of 600 ft. per minute. In descending the motor uses no current but is transformed into a dynamo generating a current which has the effect of a minus load on the engines.

The result is consequently very trying to close regulation and only the highest type of engines can be expected to successfully meet the requirements imposed by having a lighting system and electric elevator service in the same circuit.

The relative positions of the fixed and multiplying sheaves is not correctly indicated by Fig. 3, it being only a sketch giving the main and essential features of the elevator shaft, but the sectional views in Fig. 4, show the vertical heights of the different parts quite clearly.

The cable supporting the car and passing over the fixed sheave shown at the top of Fig. 3, consists of a number of small wire ropes lying side by side and each having a groove in the sheave. This method enables small flexible wire ropes to be used and a sufficient number to guarantee ample strength for any reasonable load. By reference to the sketch the ratio of 16 feet travel of the car to 1 of the nut and saddle can be readily understood. The four traveling sheaves shown are the same as the movable pulleys of a tackle-block and this number of course gives a ratio of 8 to 1. The end of the cable after passing over the last pulley in the multiplying sheaves is fastened to the bottom of the counterweight which carries the movable pulley shown attached to the top and thus multiplies the first ratio by 2 giving as a result 16 to 1. The counterweight is a frame carrying a number of castings of about the proportions indicated in the cut and the construction is such that the number can be easily changed to accommodate the requirements of the service.

The car is guided in its vertical travel by T-rails securely fastened to the sides of the elevator well and these rails will afford a good example of the displacing of shop methods in manufacture. These rails are at present finished from the rough rolled stock by special milling machines, which have a capacity for a large amount of work at a low labor cost, being so arranged that one machinist can attend two machines;

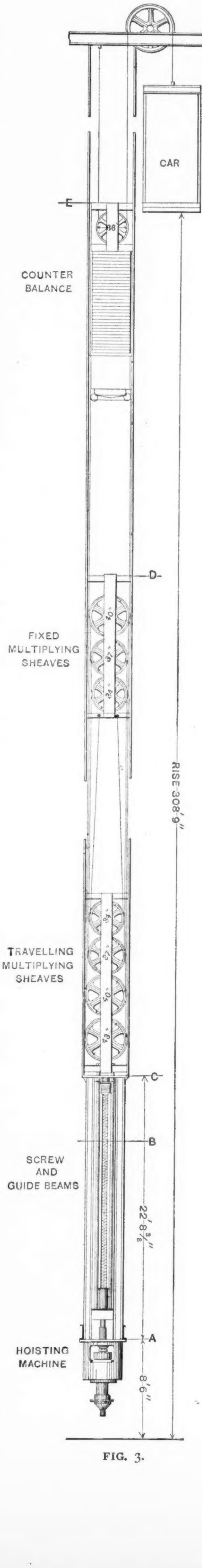


FIG. 3.

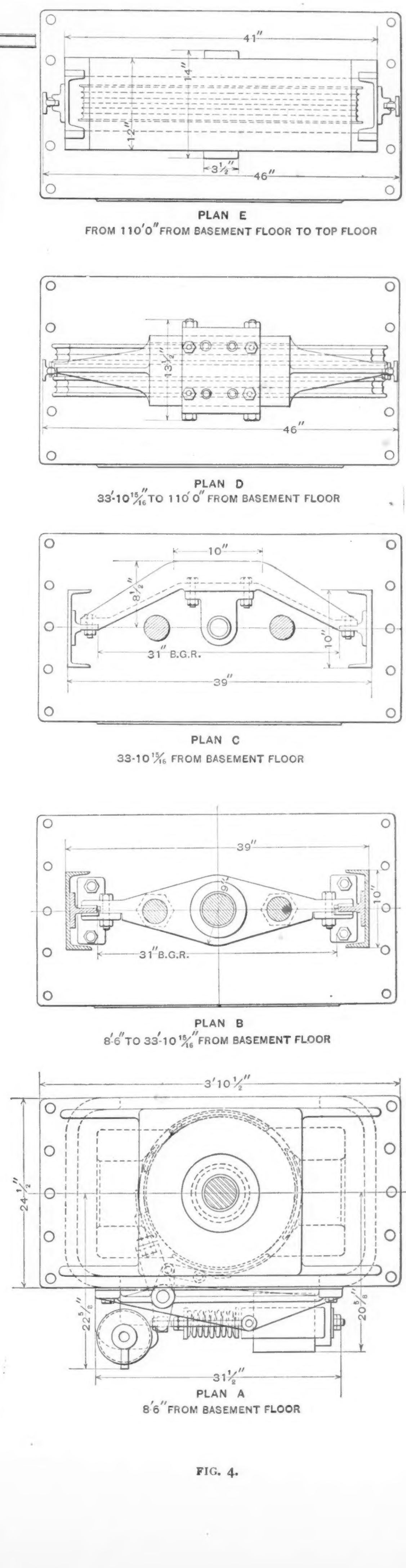
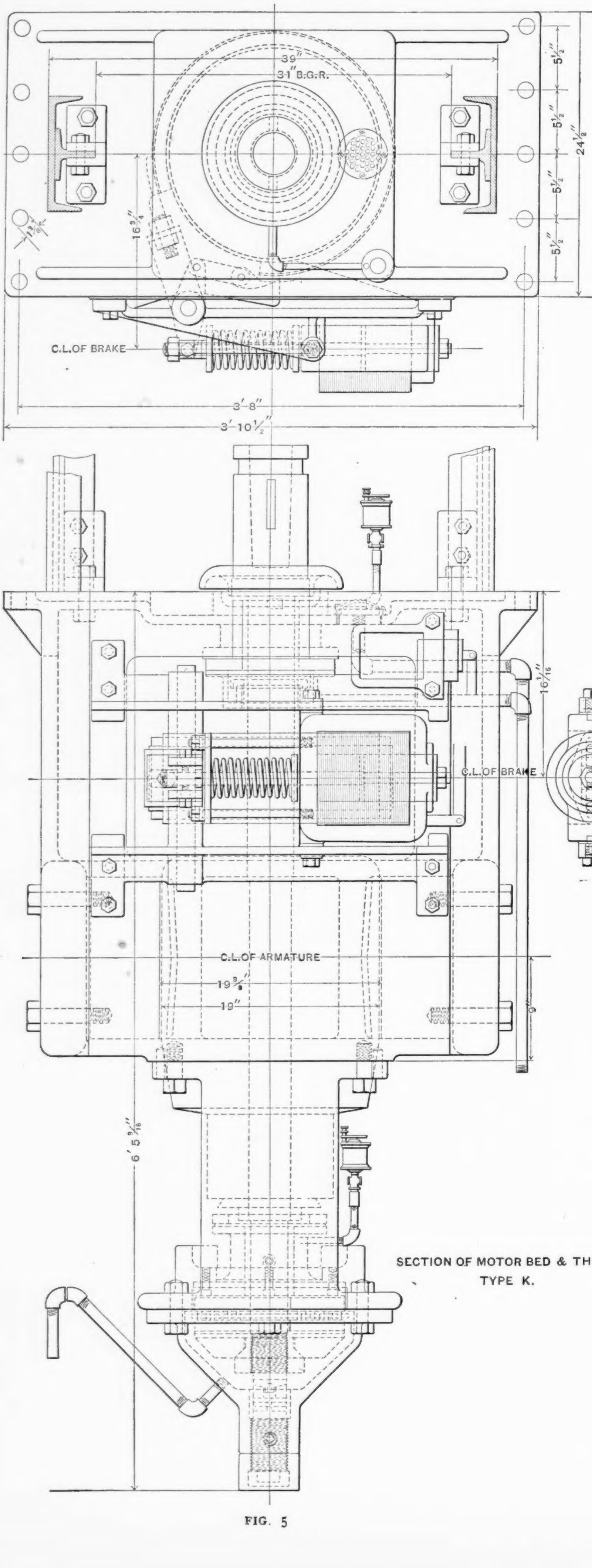


FIG. 4.



SECTION OF MOTOR BED & THRUST
TYPE K.

FIG. 5

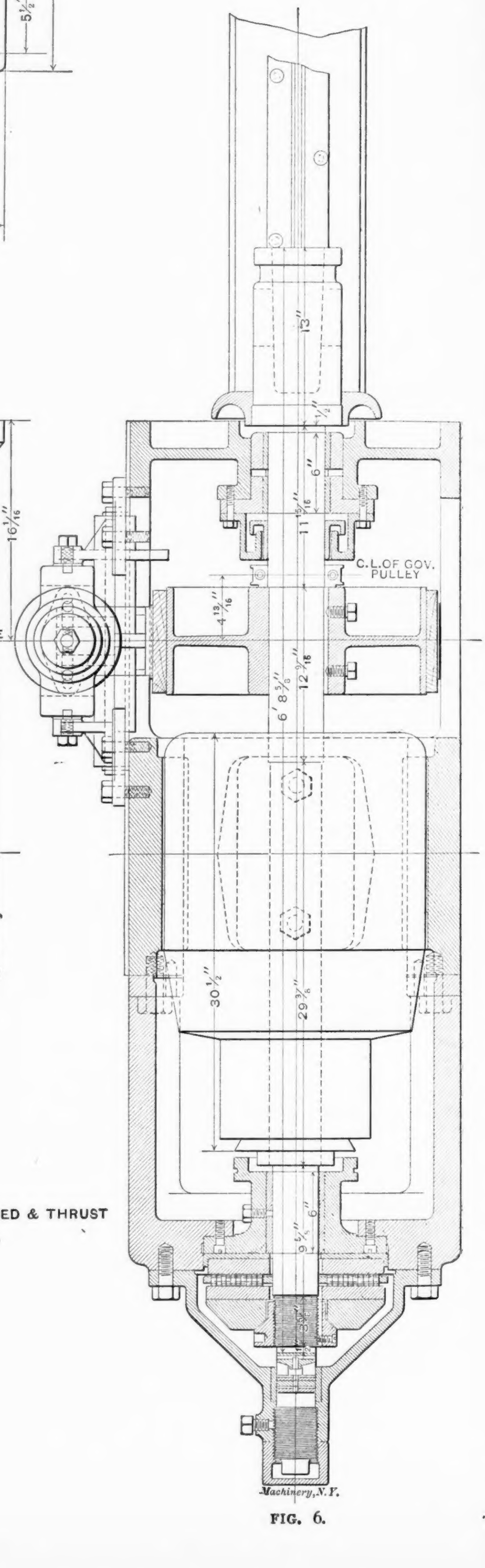


FIG. 6.

but the writer was informed that the machines were soon to be thrown out of service, as the company was now able to obtain cold-rolled stock, which would gauge more uniformly than the machined work, and cost less than the present product.

In this building, as before stated, it is necessary to have an extremely compact arrangement, and in Fig. 10 a plan of the semicircular arrangement of elevators is given, showing the elevator wells and grouping of the motors. It will be seen that the elevator mechanism is crowded into the corners, and each is indicated by the plan view of the frame and sheaves, five being shown on each side. By comparison with the other dimensions

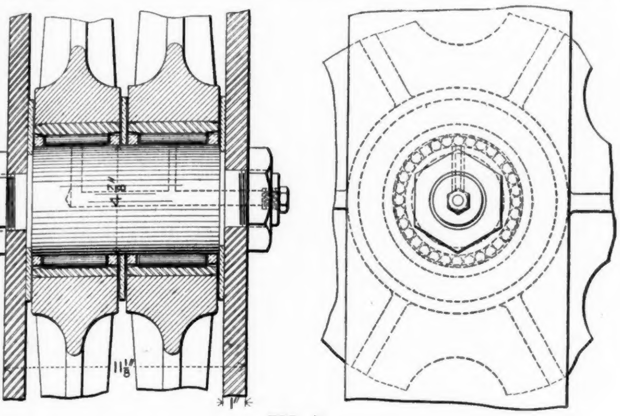


FIG. 7.

given, some idea of the constrained space can be obtained.

In all, there are ten passenger elevators, one safe hoisting machine, two tower elevators, two dumb-waiters and two sidewalk elevators, the last named being the first of their class to be equipped with a safety stop. This stop operates in case one or more of the chains supporting the platform should break, and will also go into action if the descent is too rapid, but will not operate in rising at any speed. This feature is obtained by a device similar to an engine governor, and is set so that when the centrifugal force exceeds a certain limit, the balls fly out and strike a

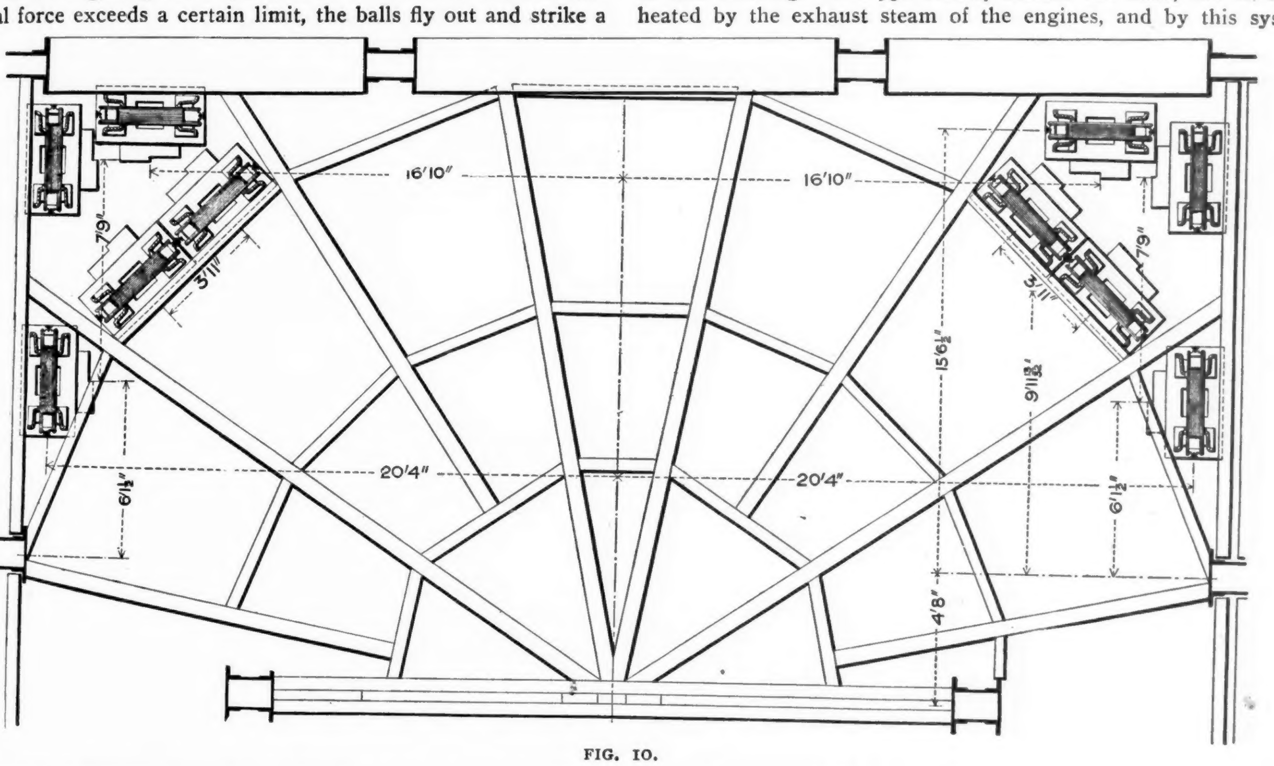


FIG. 10.

trigger that throws the grip mechanism into operation. This action, as stated, takes place if the descent is too rapid for safety, but as the ascent can be made more rapidly without danger, the governor balls will not trip the grip device on the upward travel at any speed. These are the essential features of the regular stop used in the passenger service, and it may be stated that the grip device, which was designed to seize the guide rails with a pressure, giving a factor of safety amply sufficient for the most extreme case, was found by test to require a force of two or three times the estimated amount to displace it. This result was found to be owing to the co-efficient of friction being much higher than that usually given for metals, on account of the excessive pressure per unit of surface in contact.

The power for running the elevators, the lighting service and the other apparatus requiring the electric current is furnished by five Dick & Church direct-connected engines and generators. The larger engines are rated at 1500 amperes with an electromotive force of 120 volts which would indicate about 240 horsepower. They are all equipped with the inertia governor which is so rapidly being adopted for those installations that call for the closest regulation of speed under trying changes of load.

A photograph of one of the smaller engines which is an almost

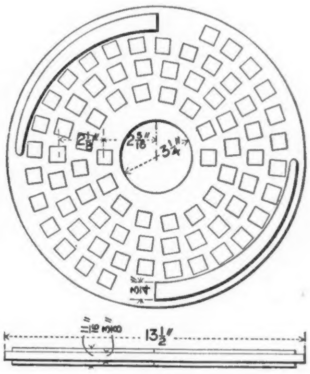


FIG. 8.

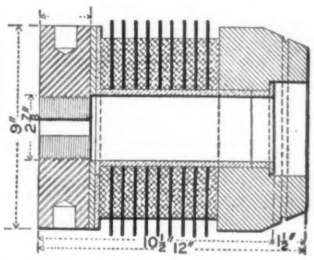


FIG. 9.

exact counterpart of the larger ones is shown in Fig. 11, but owing to conditions of the engine room at the time a picture showing the others was an impossibility. As will be noted, they are tandem-compound but not condensing as the exhaust steam is used for heating purposes.

A battery of three Babcock & Wilcox boilers furnishes steam for the engines and pumps. These pumps, which are shown in Fig. 12, are part of the Paul heating system, that is in operation in this building. The 950 offices, besides the halls, are to be heated by the exhaust steam of the engines, and by this system, when successfully operated, a vacuum of from two to three pounds below atmosphere, is maintained with ease. The system consists briefly of radiators, piping and pumps, so arranged that the radiators are at all times kept free from air and condensed water allowing the exhaust steam to rise freely to the radiators without throwing a load on the engines. Exhaust steam is practically as good for heating purposes as live steam taken directly from the boiler, and where the engines are run non-condensing, such an arrangement makes possible a heating system that costs very little for fuel maintenance.

To provide for emergencies and to allow for the most econom-

ical management of the engines a storage battery of 58 chloride accumulators is provided, an illustration of which is given in Fig. 13. These storage batteries have a capacity of 4,000 ampere hours, and with proper care are said to be very durable, but it is highly essential that the specific gravity of the liquid be kept at a uniform strength to obtain the best results.

A

B

FIG. 14. LOOKING NORTHWEST FROM THE IVINS BUILDING.

Unfortunately, the switchboard for the electric apparatus in the building, was not completed at the time our views were taken, so we are unable to present a photograph of it. This would doubtless be of interest, as, when completed, it will be one of the finest in the city, having a very complete arrangement of switches, cut-outs, etc., and being nearly forty feet in length.

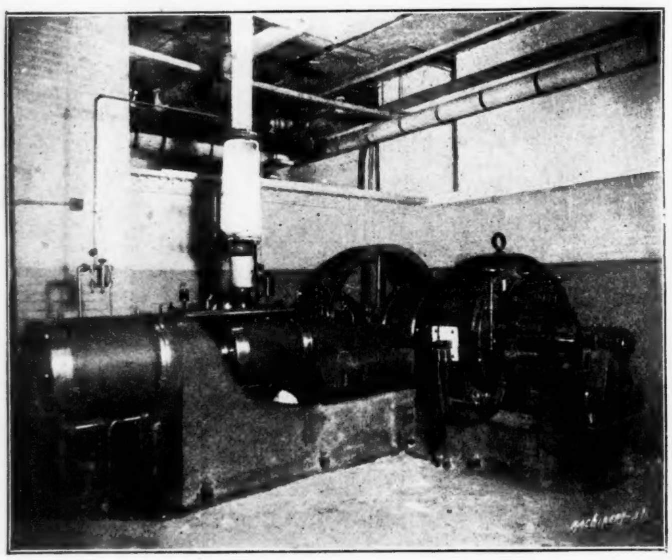


FIG. 11. ONE OF THE ENGINES AND GENERATORS.

We have been able to give but little more than a passing glimpse of the various mechanical features contained in this mammoth structure, as a detailed description of all the different features would fill the limits of this number of MACHINERY, and in closing we will touch upon a few points of general rather than technical interest.

The building is not founded on the solid rock, as would naturally be inferred, as the underlying strata precluded obtaining a foundation of this nature. The weight of 50,000 tons, which may be superimposed, is carried by a system of distributing beams upon four thousand piles, driven to such depths that each can safely carry a load of twenty tons. As a total load can be carried by all the piles equal to 80,000 tons, it is calculated by the architects that the foundation will be perfectly safe under the most extreme circumstances. It is doubtful, however whether this method of calculation is strictly reliable, where such large areas, together with excessive loads, are involved.

The two towers surmounting the building contain fine circular rooms, nearly 25 feet in diameter, and are served by elevators in their center and dumb waiters on the side. The top of the cupola of the towers is 390 feet from the street, while the flags are carried at a height of 440 feet from the earth.

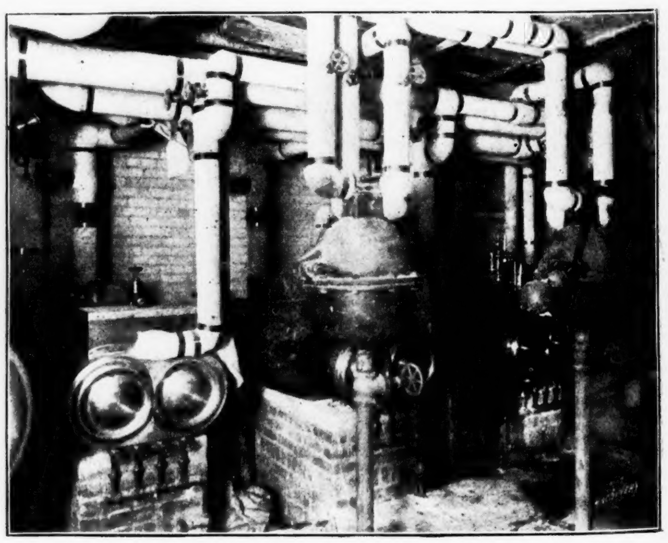


FIG. 12. A CORNER OF THE PUMP ROOM.

The main building has twenty-six stories, and the towers three which are to be occupied, making in all twenty-nine floors, besides some further up in the towers that are not to be used.

The photograph in Fig. 14 was taken from one of the upper windows, and gives a magnificent view of the city looking northwest; it also shows the home of MACHINERY, in the Franklin Building, which is the twelve-story structure in the foreground of the picture. A vertical line, from A to B, would pass directly through it. To the right, on the corner, is seen the Postal Telegraph Building, and just around the corner, on Broadway, is the Home Life Insurance Building, which was gutted by fire December 4, 1898. This office building, which was the home of "Locomotive Engineering" and until a few days before the date mentioned, had been that of the "American Machinist," was supposed to be fireproof. Considering the circumstances, however, the damage was not excessive, and probably if it had been provided with fireproof shutters, similar to those of the Ivins building, the flames would not have gained entrance. By looking closely at the cut, the rebuilding of the front wall can be seen, and the difference between the old and new marble plainly discerned.

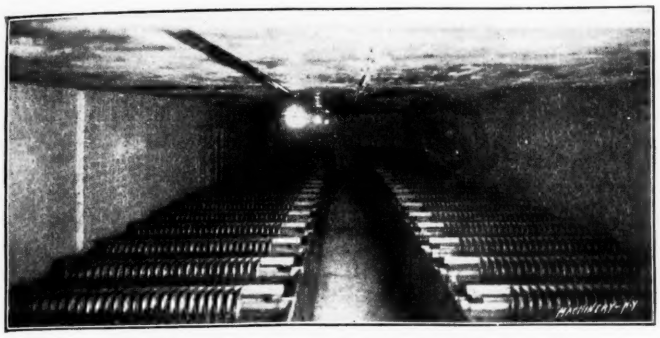


FIG. 12. THE STORAGE BATTERY ROOM.

We take pleasure in being able to present to our readers these fine specimens of the photographer's art, the reproduction on our first page being a remarkable example of the results that can be obtained by the recent improvements in photographic lenses. The negative for this picture was secured from the roof of the Astor House, and the greatest focal length obtainable was less than 250 feet, while the vertical distance clearly "cut," is nearly 500 feet.

F. E. R.

* * *

THE BECKER-BRAINARD MILLING MACHINE COMPANY.

The Brainard Milling Machine Co., of Hyde Park, Mass., and the John Becker Manufacturing Co., of Fitchburg, Mass., have consolidated under the firm name of the Becker-Brainard Milling Machine Co., which is organized with a capital of half a million dollars. Amos H. Brainard, of the Brainard Milling Machine Co., is the President and John Becker, of the Becker Milling Machine Co., is the General Manager of the new corporation. The Brainard Milling Machine Co. was organized for the purpose of manufacturing milling machines in June, 1871, through the efforts of Mr. Brainard, who was a pioneer in the manufacture of this line of machinery. The formation of this company came as the result of the need felt by Mr. Brainard for milling machines suitable for milling the parts of a bench vise which he invented in 1861, and of which many thousands were manufactured and sold by the Union Vise Co., of Boston. The Brainard Co. has made a specialty of milling machines since the date of its formation and has successfully carried on this special line of business, the Brainard machinery being known all over the world.

Mr. Becker is also one of the pioneers in the manufacture of milling machines, having been one of the first to bring the advantages of the vertical milling machine prominently before American machine builders. He made his first machine in Boston in 1889. It was a small size, intended for die sinking, and was exhibited in 1890 at the Mechanics' Fair, where it attracted a great deal of attention. In 1891 he moved to Fitchburg, Mass., where a new shop was erected and the business has constantly increased in size, the shop having been kept busy by the manufacture of this one style of milling machine, which has since been brought out in sizes suited to both heavy and light work.

The former Brainard Works were destroyed by fire a short time ago, and an entirely new shop is being built for the company. The building will be 200 feet long by 90 feet wide, and will have galleries after the modern type of shop architecture. Electric traveling cranes, and an entirely new equipment of the most modern tools will make the facilities complete and modern in every respect.

The Becker-Brainard Milling Machine Co., are to manufacture all the styles of milling machines built by the former Becker and Brainard companies, which are widely known both in this country and in Europe where the sales of the machines of both manufacture have been very extensive.

* * *

THE WORK OF THE U. S. S. VULCAN.

SOME INTERESTING FACTS ABOUT THE REMARKABLE WORK DONE BY OUR FLOATING MACHINE SHOP.

An informal reunion of the junior members of the American Society of Mechanical Engineers was held at the house of the society on the evening of March 7. About seventy-five juniors and seniors were present, and an entertaining lecture was given jointly by Chief Engineer Gardiner C. Sims and Assistant Engineer William S. Aldrich, of the U. S. N. repair ship Vulcan, upon the experiences of those who served on this vessel. Mr. Sims was the first speaker, and gave a thrilling and a deeply interesting account of the condition of the Maria Teresa and Hobson's work in raising her; of her trip North in tow of the Vulcan, and of her abandonment at sea during a hurricane, of the work of saving the American crew that had been placed on board of her, and finally, of the stranding of the vessel on Cat Island, which, the speaker said, had been identified as the very island on which Columbus first raised the flag of Spain.

Assistant Engineer Aldrich, who, in times of peace is in charge of the mechanical and electrical departments of the University at Morgantown, West Virginia, followed Mr. Sims, and confined his remarks mainly to the equipment of the Vulcan and the work done while stationed at Guantanamo Bay, with brief sketches of the life led by the mechanics and members of the crew of the "floating machine shop" during its long and trying stay on the coast of Cuba.

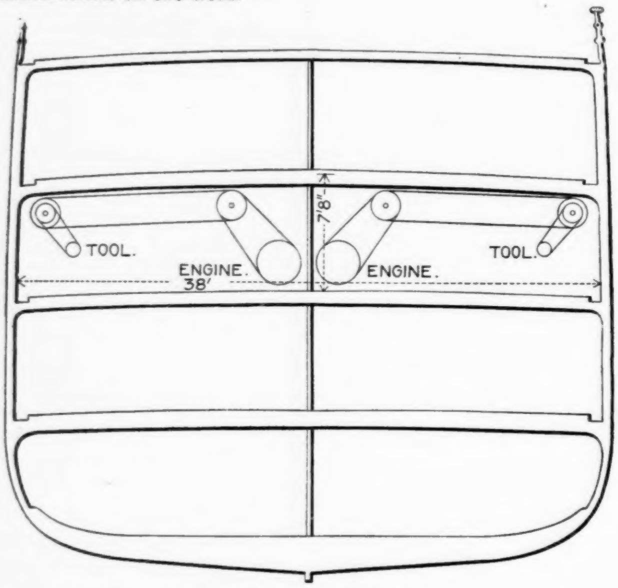
The accounts of both speakers were illustrated with many lantern slides, which added much to their general interest.

In what follows, we give a few facts, presented by Prof. Aldrich, relative to the work of the Vulcan during the months that the ship was with Admiral Sampson's fleet, together with a few additional points, kindly furnished us by him subsequent to the lecture.

The Vulcan was formerly the steamship Chatham, of about 3,000 tons displacement, and was well adapted for her work. She was fitted out last June at the Boston Navy Yard, and on July 1 reported to Admiral Sampson, off Santiago, and after offering her services as a combatant, was remanded to the naval base so gloriously established by our marines at Guantanamo Bay, about forty miles distant from Santiago. The Vulcan's honors were not to be won with Sampson's fleet, in battle, but in repairing his fleet after the battle. Its picked crew of skilled mechanics, comprising machinists, pattern-makers, moulders, blacksmiths, boiler-makers, coppersmiths and others, and her great quantity of engineers' stores, were all designed for the work of construction, and how well she accomplished the purpose for which she was designed, the enumeration of some of the repairs which her force were called upon to make, is the best of evidence.

Through the courtesy of Commodore Melville, who furnished us with a blue print of the Vulcan, we are able to show the accompanying sketch, indicating the arrangement of the shafting, belting and tools of the machine-shop deck. The inconvenience resulting from the little head room can better be imagined than described. The belts were so low that one had to duck under them in moving about, and found his hair continually rising, not from fright, as might be expected, but from the frictional electricity generated by the belts. The adjusting bolts of the hangers, too, continually brought the passer-by to order, and sharply reminded him of their presence. The largest lathe was of 40-inch swing, and there were a dozen others of smaller size. There was a 48-inch radial drill, a planer 4 x 4 x 10 feet, and a full equipment of shapers, drill presses, pipe-threading machines, small tools, attachments, etc. Some of the machines were ranged around the sides of the deck, and others were placed near the center. The feature of this deck, however, as well as the most

remarkable arrangement for repair work ever placed on shipboard, was the foundry, equipped with a 36-inch four-ton iron cupola and brass furnaces. The foundry floor was 16 feet square, of cement, and the deck overhead and the side walls were protected by asbestos, sheet metal and sand; yet it was almost impossible to prevent a blaze when pouring off, the heat was so intense in the narrow quarters. When the cupola was dumped, men stood ready with the hose, and on three different occasions the woodwork caught fire, which was not altogether pleasant, in view of the fact that the Admiral had stored tons of gun-cotton and loaded shells in the hold.



SECTION OF THE VULCAN.

The skilled mechanics numbered about 100 men, who supposed that they were going to assist in making repairs to the machinery, boilers and piping of the victorious fleet. This they had to do, but they found, also, that the Vulcan had become a floating shipbuilder, a manufacturing plant, a source of supply from which men were drawn for every variety of outside work, including the resuscitating of certain vessels of the Spanish fleet. They had to take their turns at watch, and thus had a foretaste of what the new Navy Personnel bill will be if it becomes a law. No sooner had the Vulcan come to anchor for her three months' stay in the tropical climate, at Guantanamo Bay, than the rush of orders commenced, and it seemed to those on board that about everything on every ship of the fleet had given way at once. Never had the indolent Cubans witnessed such industry by day and much less by the glowing fires and bright lights of the night. Admiral Sampson himself became alarmed at this aspect of the night work, thinking the brilliantly illuminated ship afforded too good a target for the Spaniards in the surrounding hills, and "lights out" was the order. But it soon became evident that the necessary repairs could not be made without working overtime, and the order was rescinded.

It should be noted that the so-called repair work included not only work coming strictly under this head, but the making of entirely new parts, including the patterns, castings and forgings, and sheet metal work. Parts of vessels damaged by shells were repaired, including the making of a new battle-hatch for the Iowa. The Oregon, be it said to her credit, did not ask for repairs. Apparently, Engineer Milligan and his junior officers had the faculty of keeping everything in order, under even the most trying conditions. One requisition that was made, however, and complied with, was for seven or eight spare plug cocks for the Oregon, to be carried in stock! Engineer Milligan made it a rule to carry extra parts in stock to guard against emergency, and his calls upon the Vulcan were solely for work of this description.

Some of the more important and frequent repairs that were made are thus enumerated by Prof. Aldrich:

"Reversing or jacking engines had broken worms, bent shafts, loose and worn collars, keys out of shape, and required general tightening up all around.

"Capstan engines presented broken cast iron cross-heads, worn wrist-pins and very many broken eccentric straps, and badly worn boxes requiring new ones or re-babbitting old ones.

"Winch engines required the same line of repairs as capstan engines, especially in worn boxes and broken eccentric straps.

"Crane engines required new boxes, new worm wheels, new shafts, new keys and much general overhauling. The cast worm wheels were particularly weak, and where not broken in two parts, came up for repairs with several teeth missing.

"Blower engines had broken cylinder heads and required new pistons and packing rings and overhauling of slipper slide.

"Steering engines required chiefly a general overhauling.

"Dynamo engines presented broken mild cast-steel cross-heads, broken eccentric straps and rods, where cast in one piece, bent and broken valve stems, broken cylinder heads, worn piston valves and loose wrist pins. One governor wheel required to be rebushed.

"Dynamos were tested for grounds and broken circuits in fields and armatures, and one commutator was trued up.

"Electric motors attached to the small blowers were overhauled and the armatures of two of them rewound.

"Ice machines required new valve stems, slide valves and cross-heads.

"Steam pumps called for most numerous repairs to the valve motion, and many of the duplex type required rebushing throughout. Broken slide valves and supplementary valves were replaced by new ones. The water ends evidently gave the most trouble, and required most general overhauling, chiefly in the soft packing and water pistons. The small duplex pumps of steam launches were continually getting out of order.

"Steam boilers required scaling, new baffle plates, new grate bars, new internal feed systems, new tubes and expanding and rolling old ones; replacing of badly corroded connections on account of not using zincs properly, repairs to blow valves on account of deranged or broken parts, new tube stoppers, manhole dogs and bolts; calking, particularly of bottom seams; and jacking up of furnaces, down in the corrugation or flat (cylindrical) surface or both, in the same furnace. Launch boiler required new dome and ash pan.

"Globe and gate valves and plug cocks required reseating, new valves, new top and stems, new plugs. In many cases old valves were entirely replaced by new valves taken from engineers' stores or made from patterns and castings finished in the machine shop. Air pump valve seats, studs and nuts were in demand.

"Many manufacturing orders were executed for spare parts, such as air pump valve seats, ash hoists, castings, plug cocks, wrist-pins of small engines, special studs and screws, special spanner and socket wrenches, reducing hose couplings, crank and ring flanges to check valves, valve stems and bushings."

In an article published in the Athenaeum, a college publication of Morgantown University, Prof. Aldrich thus relates how the closing days of the stay at Guantanamo Bay were spent:

"Our Decoration Day was Sunday, September 4, when almost the entire crew of the good ship Vulcan went ashore to decorate the graves of the marines and sailors buried on Camp McCalla, the name now given to the spot where the old Spanish blockhouse stood. The following day being Labor Day throughout the States, was properly observed by our engineers and mechanics by the first all-American baseball game on Cuban soil. The game of war seemed long since past. The games of peace engrossed our attention. Our first practice game enabled us to win a decided victory over the Scorpion's crew about a week later. Score: Vulcan, 18; Scorpion, 14.

"After doing about all the work that was necessary for fifty-four vessels of Sampson's fleet and that of the eastern squadron, commanded by Commodore Watson, we turned our attention to the repairing of the Spanish navy. The U. S. S. Marblehead raised the Spanish gunboat Sandoval, that had been sunk by her treacherous captain off Caimanera, after the surrender of Linares. The Vulcan made two trips to the wrecks lying on the beach to the westward of Santiago harbor, and assisted in convoying the Maria Teresa from her last resting place to a safe anchorage in Guantanamo Bay. When the Maria Teresa came off the beach amid the blowing of steam whistles and successive firing of national salutes of twenty-one guns, every one felt that she had been resurrected unto a new lease of life, destined for a career in a navy from whose officers and men she would receive the most considerate treatment, and never again be headed to shore under an enemy's fire or ignominiously stranded on the softest spot on the beach."

STEAM POWER OF SMALL YACHTS.

PLAIN RULES FOR CALCULATING THE DISPLACEMENT, HORSE-POWER, SPEED, AND THE NECESSARY POINTS ABOUT THE PROPELLER FOR SMALL BOATS OF MEDIUM SPEED.

William Burlingham.

There may be data published upon the powering of small boats, but if so, it is as difficult to find as the proverbial needle in a haystack. The following is an attempt at providing a means for determining the necessary points involved in this work:

It is designed to suit the needs of amateur and practical small boat builders, who have not had the opportunity to acquire the necessary mathematical knowledge, requisite for this designing.

A little experience combined with the graphical tables is highly desirable, for the author fully recognizes the difficulty of laying down hard and fast rules for the design of propellers and the powering of boats, still it is quite necessary to have something to start on.

In laying down the lines for a boat of this small type, it is necessary to consider, whether we are to use her as a barge or a moderately fast and comfortable boat. The data in this article are designed to meet the requirements of the latter type—that is, a boat fit for cruising and general utility work.

The Lines of a Fast Boat.

A general description of the lines of the Norwood, designed by Mr. Mosher, would not be out of place here, as showing the general trend of the design, modified of course, for a boat of this description.

Above water, the Norwood has a plumb stern and straight sheer with her sides nearly vertical. All the level lines from deck to water are round and full, with greatest breadth well aft, and carried out with little diminution to the extreme stern. The floor is nearly flat, carried well fore and aft, with a short round to meet the sides. At its after end, over the propeller, it is concave, similar in shape to the bowl of a spoon. The edges of this spoonlike surface join the sides at a sharp angle.

The screw should be located in such a position as to obtain a full and free supply of water and also a free discharge as it leaves the propeller. If the water is obstructed in its discharge, it is apt to be carried around with the revolving propeller to the detriment of the propulsive effect.

With these few words regarding the most desirable shape for a boat we will proceed to take up the powering of the same.

Rule for Displacement.

One of the first things necessary to know is the displacement of the boat we are to power. As the rules and formulæ for finding this are tedious for one unaccustomed to the work, we will approximate them by the formula using a constant block co-efficient of fineness. As experience accumulates this co-efficient may be varied to suit the different types.

By the block co-efficient of fineness, is meant the ratio between the displacement volume of the boat, and a rectangular block with a length, breadth and thickness equal respectively to the water-line, length and beam, and the draft, mean, of the boat.

A navy launch with a carrying capacity of ten men, arms and ammunition, has a coefficient of about .35, and a speed of from 8 to 10 knots. A high speed launch or yacht, has a co-efficient of from .2 to .25, so for our boat we will take a mean of .3 for a moderate speed and comfortable quarters.

The formula then is:

$$\text{Displacement volume in tons} = \frac{.3 \times L \times B \times D}{35}$$

Where L = length at water line.
B = beam at water line.
D = mean draft.

Indicated Horse-Power and Speed.

The displacement being thus approximately found, the curves on sheet A will give the indicated horse-power, and corresponding speed for boats of from one to fifteen tons. In case the tonnage is intermediate, between the curves shown, it may be found by interpolation, dividing the intermediate spaces between the curves, by spacing on the dotted lines A.A.

By locating the speed and power points of reliable trial data of similar boats upon this map of curves, we will eventually have data giving the extreme variation from the curves shown and in designing new boats will be able to allow a certain per cent. above or below the curves for our power, according as the average power data is above or below.

This sheet of curves was laid down as follows:

From the trial data of a number of boats of a displacement approximating those shown by Froude's Law of Comparison, they were reduced to a similar boat of seven tons displacement, with corresponding powers and speeds. Spotting the points thus

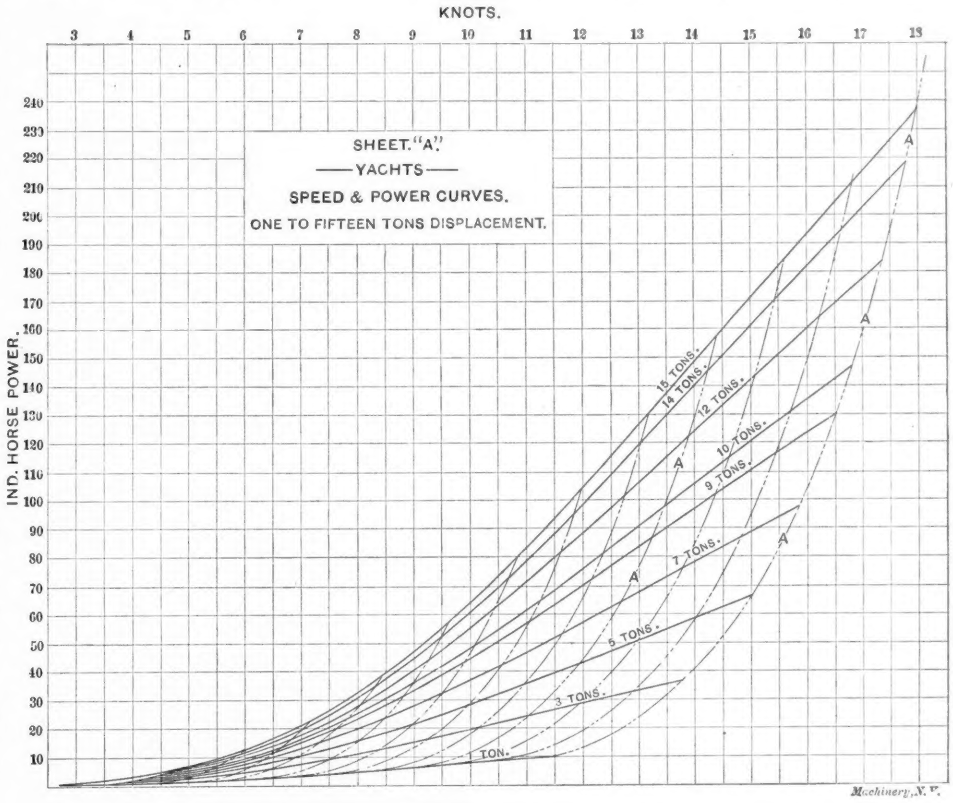


Diagram showing the required horse-power for different speeds and tonnages. Dotted lines A A are merely for guides in plotting intermediate points, as explained in the text.

obtained, an average speed and power curve was drawn in. By the same law, from the points obtained by this latter curve of seven tons, a comparison was made for boats of from one to fifteen tons, and these points spotted and the curves drawn.

Froude's Law of Comparison is used for finding the speed and power of similar ships of different displacements, the speed and power of one being known, and the formula is thus expressed:

$$\sqrt[6]{\frac{D_1}{D}} = R \qquad \frac{D_1}{D} \times R = S$$

D_1 = displacement required.
D = actual displacement.
R = speed constant } for the required displacement.
S = horse-power constant. }

Multiply the speed and power of the boat of "D" displacement by R and S, respectively, and we will have the speed and power of a similar boat of D_1 displacement.

Dimensions of the Engine.

The required horse-power having been found, we must now determine the size of the engine necessary for furnishing this power. A convenient set of formulæ are the following:

$$\text{I. H. P.} = \frac{d^2 \times p \times S}{42000} \qquad \text{M. E. P.} = \frac{\text{I. H. P} \times 42000}{d^2 \times S}$$

$$d^2 = \frac{\text{I. H. P.} \times 42000}{P \times S}$$

Where d = diameter of cylinder in inches.
p = mean pressure in lbs. per square inch.
s = speed of piston in feet per minute.

The piston speed of high speed yachts varies usually from 800 to 1000 feet per minute, but for the type of boats that we are figuring on, a speed of 700 feet is probably the best.

The mean pressure referred to the low pressure cylinder, would be about 35 pounds, and this may be used in the formulæ. Some of the Mosher and Seabury boats will run as high as 60 pounds, but for the average engine that we can afford to build, 35 pounds is enough. The proportion of cylinder volumes for a triple expansion engine using steam at 180 pounds pressure would be as 1 to 2.54 to 7.6.

For a compound engine, with steam from 120 to 160 pounds, the ratios of cylinder would be, as 1 to 3.75 or 4. A two cylinder simple engine is often used with success in this type of boat with steam of moderate pressure, from 120 to 140 pounds.

In these days of small, high-pressure, tubulous yacht boilers, a boiler pressure of 250 pounds is advisable, it offers the advantages of less weight and less space, very desirable features in a small boat. With a reducing valve at the engine any desired pressure may be obtained. Many torpedo boats use this reducing value and there seems to be no valid reason why it would not work well on a small boat. It would give plenty of good dry steam, and does not complicate the pipe arrangement at all.

It is impossible, in an article of this length to go into a detailed description of the best types of engine. There are many such, with widely varying details. To secure the best results, the best materials should be used; as the weight is so small there is no excuse for not using the best grades of steel and alloys. Steel is procurable in open market of from 80,000 to 95,000 pounds tensile strength, and should be used for the more important parts of such engines.

The 80,000 pound steel is nickel steel.

The 95,000 pound steel is nickel steel, oil tempered and annealed.

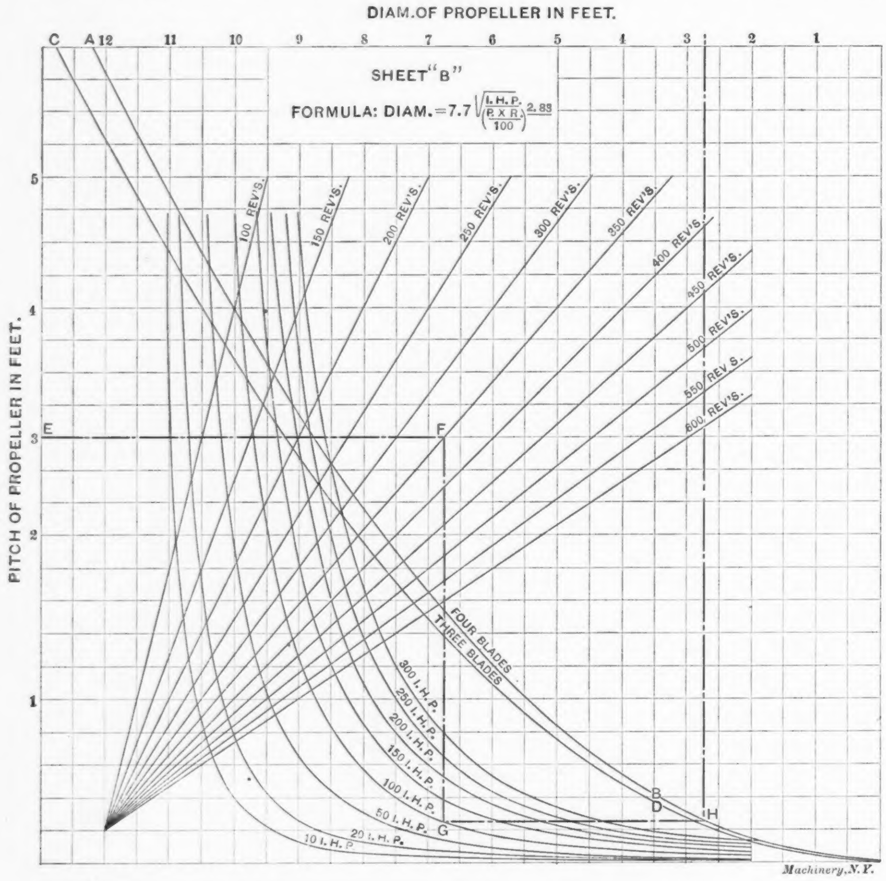


Diagram for finding the diameter of a propeller for a given pitch, the number of revolutions and horse-power being known.

The piston speed and horse-power having been found, the revolutions depend upon the stroke of the engine, and this must be fixed before deciding upon the propeller.

The Propeller.

Two-bladed propellers are now seldom used on account of the vibration set up by the varying moments of the thrust reactions.

Three-bladed propellers work very well when sufficiently immersed and with a fairly high number of revolutions, and are very often used for small boats with success. But the propeller that suits this type of boat best is the four-bladed, as the draft of water is usually light. They are better balanced and reduce the vibration of the boat.

They are usually cast in one piece of composition, made of 88 per cent. copper, 10 per cent. tin, and 2 per cent. zinc, or of manganeze bronze. The latter is the best as the blades may be cast thinner, the comparative strength of the two metals being as 32 or 33,000 is to 58 or 60,000.

We can take the average slip of the propeller as 20 per cent., and the pitch may then be found by the following formula:

$$\text{Pitch} = \frac{S \times 10133}{R \times (100 - x)}$$

Where S = speed of boat in knots per hour.
R = revolutions of propeller per minute.
x = slip (20 per cent.).

A knot is 6080.27 ft. in the United States, usually called 6080 ft., and is equal to one-sixteenth part of the length of a degree on a great circle of a sphere, the surface of which is considered equal to the surface of the earth.

On sheet B is shown a graphical method for finding the necessary diameter of a propeller, and is laid down from the formula.

$$\text{Diameter} = 7.7 \sqrt{\frac{\text{I. H. P.}}{\left(\frac{P \times R}{100}\right)^{2.83}}}$$

for four-bladed screws.

The above diameter multiplied by 1.05 for three-bladed screws. Curve A. B. gives the diameters for the four-bladed, and curve C. D. the diameter for the three-bladed propellers.

The method of using the curves is as follows: Suppose we have a pitch of 3 ft., revolutions 350, and horse-power 100. Have the course shown by line E. F. G. H. I, thus: From the pitch run a line horizontally to the oblique line representing the revolutions, then vertically downwards until the desired horse-power is reached, then over horizontally until the curve A. B. is reached, if a four-bladed propeller is desired, to C. D, if a three-blade, then vertically upwards to the top horizontal line when the required diameter will be found in feet and decimals of a foot.

On Sheet C, will be found the developed area, corresponding to the diameter and pitch found. The curves were constructed form the formula.

$$\text{Developed area} = C \times \frac{\sqrt{\text{I H. P.}}}{R}$$

Where C = constant 7.
R = revolutions per minute.

The method of using these curves is as follows:

Supposing we have 450 revolutions and 100 horse-power. Then from 450 revolutions proceed vertically upwards to the 100 horse-power curve, then horizontally to the curve AB., and then vertically upward to the top horizontal line, where the developed area will be found in feet and tenths of a foot. The dotted line C. D. E. F., shows the course to take.

Shape of Propeller Blade.

We now have the displacement, tonnage, indicated horse-power, speed and diameter, pitch and developed area of pro-

peller. The shape of the blade is the next consideration. The shape given in U. S. Naval Constructor, D. W. Taylor's book is as good as any, and easy to lay down.

The diameter of the hub is taken as 2-9 of the total diameter of the propeller, but this may be slightly changed to suit each case, without changing the characteristics of the blade.

The diameter being settled, then

$$\text{Width of blade} = \frac{\text{developed area}}{.35 \times \text{diameter}}$$

About the Boiler.

Regarding the boiler necessary I should say that the three best were the Ward, Mosher, and Seabury. These boilers are compact, good steamers, comparatively easy of repair, long lived, and fairly safe from explosion.

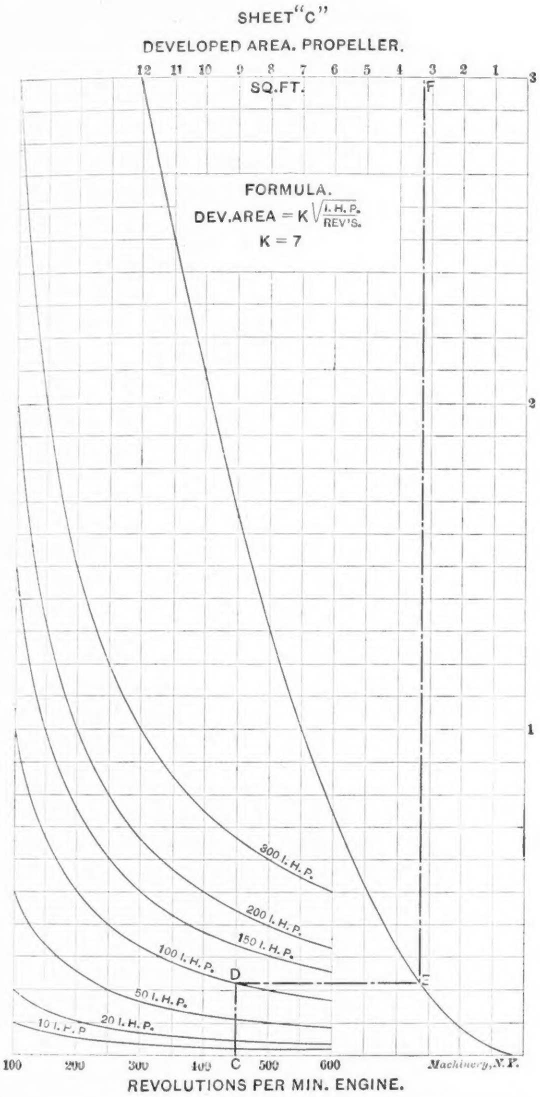


Diagram showing the developed area of the propeller corresponding to the pitch and diameter found.

They can be bought in very small sizes as cheaply as can be made without special tools. The only points regarding these boilers that concern this article, is the size necessary. We cannot figure on more than 20 one-horse power per square foot, of grate for these small boats, and the ratio of the heating surface to the grate surface should be about 35 to 1.

These boilers must be fed with fresh water, and if the boats are to be run on the salt seas, etc., they should have ample tankage capacity for fresh water.

A boat using triple or compound engines must of necessity have condensers. The best type for these boats is the long copper pipe keel condenser. The Bailey or Edwards air pump will give the most satisfactory results; they are worked from an eccentric on the main shaft.

An independent feed pump of the small size is better than one worked from the engine, in fact, it is almost a necessity with these boilers. Beneath the boiler is usually placed a flat feed tank, part of the regular tankage, and it extends beyond the front of the furnace, forming a sort of hearth. The various details of engines, boilers, etc., do not come within the scope of this article, and cannot be touched upon, for they are really another story.

* * *

RULES FOR DETERMINING THE HORSE-POWER OF BELTS.*

A PRACTICAL DISCUSSION OF THE BELTING PROBLEM, WITH RULES AND TABLES FOR MAKING ALL NECESSARY CALCULATIONS.

The problems in belting, although apparently simple, are in reality so complicated, and so many considerations are involved in their solution, that we are not surprised at the multiplicity of varying rules and formulæ by which the power transmitted by a given belt may be obtained.

There are in general five principal considerations which should obtain in determining the size of belt, viz., co-efficient of friction, arc of contact, speed of belt, strength of belt per square inch of section and durability.

Experiments on Belting.

Experimental work has usually been directed to determine the co-efficient of friction, and it has been shown by exhaustive tests that the co-efficient of friction Θ, of leather belting on smooth cast-iron pulleys may vary from 12 per cent. to 165 per cent., depending upon so many factors that it is impossible to so corelate them as to determine accurately what the co-efficient may be in any given case.

In general we may say that the value of the co-efficient depends upon the nature and condition of the leather, the percentage of slip, temperature and humidity, and probably pressure.

The velocity of sliding, which influences the friction more than any other factor, is directly proportional to the belt speed, and may safely be estimated at one per cent of the speed per pulley, that is two per cent. for each pair of pulleys. The maximum slip permissible with horizontal belts is about 20 per cent.; a greater slip than this is liable to throw the belt off.

For the past 25 years the co-efficient of Towne & Briggs, viz., 42 per cent. has generally been accepted by engineers, but this was deduced from the results of experiments having a large slip (200 feet per minute), which gave the co-efficient equal to 0.58, so that the value recommended is only 0.70 of that obtained by experiment. Even with this allowance it is probable that the value is somewhat higher than average practice would warrant. With percentages of slip varying from 0.8 to two per cent., Mr. Lewis finds the co-efficient to vary from 0.25 to 1.38, the first value being obtained for a percentage of slip equal to 1.5, and the second for a slip of 1.7 per cent. Two per cent. slip gave $\Theta = 0.45$. Other experimenters find that Θ varies from 0.12 to 0.58, depending principally upon the rate of slip.

Prof. Lanza's numerous tests, made at the Massachusetts Institute of Technology, indicate that 27 per cent. is the most suitable value to use for a low rate of slip. His results were obtained on belts running about 1,500 feet per minute and with a slip of about three feet per minute.

Regarding the strength of belting, as might be expected this is extremely variable. Kirkaldy's tests, which cover a wide range, show the strength to vary from 2,000 to 6,000 pounds per square inch of section.

The various tests indicate that double belts are superior to single, both in respect to their greater uniformity and also a lesser liability to become crooked by reason of uneven stretching, due, probably, to the fact that the former are composed of two layers, which tend to neutralize any local imperfections occurring in either. It is evident that the strength of a belt cannot be stronger than that of the joint. With cemented joints the splice, if properly made, is not materially weaker than the body of the belt; with riveted joints the ultimate strength, as given by Towne, is 1,750 pounds per square inch, and for laced belts 960 pounds.

Prof. Walter Flint, of Maine State College, finds that metal belt fasteners may cause the joints to fail at from one-fourth to

* Abstract of a paper read by John J. Flather before the North-west Railway Club.

one-half the strength of solid belt, and that laced joints may rupture under a stress of 30 per cent. to 60 per cent. strength of solid belt.

In existing belt transmissions the stress per square inch of section is extremely variable, ranging from 30 to 1,200 pounds per square inch. The reasons for this are two-fold. First, a superior quality of leather will permit a greater stress without undue trouble and expense in repairs; and secondly, some belts are calculated to transmit a stress greatly in excess of that to which they are actually subjected.

Durability of the Belt.

A feature often neglected in the ordinary formulæ is that of durability of the belts. F. W. Taylor, in his "Notes on Belting," states that "the one consideration which should have more weight than all others in making up tables and rules for the use and care of belting is how to secure the least possible interruption to manufacture from this source."

In calculating the total expense of belting and the manufacturing cost chargeable to this account, it has been observed that by far the largest item is the time lost on the machines while the belts are being relaced and repaired; especially is this true in those establishments where the running of one series of machines is dependent one upon another, and the stoppage of one machine involves delays on others.

The results of Mr. Taylor's observations on belting, covering a period of nine years, constitute a valuable compendium on the subject and should be carefully studied. Among many interesting conclusions we note the following:

(a) The belt speed for maximum economy should be from 4,000 to 4,500 feet per minute.

(b) Belts are more durable and work more satisfactorily made narrow and thick rather than wide and thin.

(c) It is advisable to use double belts on pulleys 12 inches in diameter or larger, and triple belts on pulleys 20 inches in diameter or larger.

(d) All belts should be cemented instead of laced or fastened in any other way.

(e) When a belt is spliced the tension used should be ascertained by means of a spring balance connected to belt clamps.

(f) The total life of belting, cost of maintenance and repairs, and the interruption to manufacture caused by belts are dependent upon the total load to which the belts are subjected more than upon any other condition. The other conditions are: Method of splicing, care in properly greasing and keeping them clean and free from machine oil, and the speed at which they run. This latter has little effect under 2,500 feet per minute.

(g) The total stretch of leather belting certainly exceeds six per cent. of its original length.

(h) Oak-tanned and fulled belts are superior in all respects (except co-efficient of friction) to either the oak-tanned not fulled, the semi-rawhide or rawhide with tanned face.

(i) If double leather belts are tightened while at rest 71 pounds per inch of width, and subjected to an additional working load of 65 pounds per inch of width, their tension will fall in 2½ months so as to be, while at rest, 33 pounds per inch width, or 106 pounds per square inch section; their average tension during these 2½ months being 46 pounds per square inch width, or 150 pounds per square inch section; their average load during these 2½ months being 111 pounds per inch width, or 358 pounds per square inch of section. These are the conditions under which belts work when tightened according to the ordinary rules.

(j) The most economical average total load for double belting is 65 to 73 pounds per inch of width; i. e., 200 to 225 pounds per square inch of section.

(k) A double belt having an arc of contact of 180 degrees will give an effective pull of 35 pounds per inch width for an oak-tanned fulled belt, and 30 pounds for other leather belts and six to seven-ply rubber belts; or 950 feet per minute of one inch wide oak-tanned and fulled belt will transmit one horse-power, and 1,100 feet per minute of other leather belting and six to seven-ply rubber.

(l) Belts should be cleaned and greased every five to six months.

A glance at some of the existing belt formulæ shows that the older millwrights frequently allowed 550 feet per minute for a single belt, only 25 per cent. of that given in section (k), for ordinary belting.

The probable reason for the satisfactory running of these belts has already been pointed out, viz., overrating the capacity of the machine and estimating upon a horse power three or four times greater than actually required.

Some of the more recent rules used by shopmen assume that a single belt one inch wide will transmit one horse-power for every 800 to 1,100 feet per minute; Mr. Taylor's rule, it will be noted, calls for a double belt of the same width.

While the conclusions arrived at by Mr. Taylor are extremely valuable from an engineering as well as commercial standpoint, the first cost of equipping a plant under his rules would often prohibit the use of such heavy belts; moreover, in many situations it is probable that equally satisfactory results are obtained under stresses twice as great as those given, especially in those cases where the belts runs at moderate speed over a pair of approximately equal pulleys of large diameter, and is not shifted from one pulley to another. On the other hand, the conditions might be such as to call for certain belts fully as wide if not wider than those given by Mr. Taylor's rules.

Such a case might arise when running at a high belt speed from a large to a very small pulley (or vice versa) with the tight side of the belt on top; in this case the centrifugal force would diminish the effective tension, the arc of contact would be lessened, and the adhesion of the thick belt on the small pulley would not be as effective owing to imperfect contact, due both to air entrainment and to a wrinkled belt produced by bending; all of these would necessitate an increased width of belt.

Owing to these various causes, it is impracticable to determine a general formulae which will be applicable in all cases. We may, however, suggest limitations. As some one has aptly remarked, a significant factor (which, however, cannot enter algebraically into the equation) is the judgment of the designer.

Usually the conditions are such that the principal modifying agencies are known, and if these be taken into account by choosing suitable constants (to be selected for any particular case when the data are known), a proper belt width may be obtained which it is believed will be entirely satisfactory, considering durability and expense of operation under those conditions which are liable to arise in ordinary practice.

Belting Formulas.

For the purposes of the present paper it will be unnecessary to enter into a detailed mathematical discussion of the derivation of the formulae. It will be sufficient to state that the driving force P in any case is equal to the difference of tension T_1 on the driving side of the belt T_2 on the slack side, and noting that the driving force must equal the friction between the surfaces, we obtain $P = T_1 - T_2 = F$.

The friction F, depends upon the arc of contact, between the belt and pulley, the co-efficient, between the surfaces in contact, and upon the centrifugal force F°, set up in the belt, due to its velocity and weight; it is, however, independent of the diameter of pulley. To determine the values of F, T_1 and T_2, it will be necessary to assume a working tension, f, in the belt; also its speed and weight, co-efficient of friction and arc of contact.

If we assume that the working tension per square inch of a leather belt (cemented joint) equals 350 pounds, a convenient value will be obtained for f and one which will give satisfactory results when the belts are properly cared for.

For laced joints three-fourths of this value may be used, i. e., 265 pounds per square inch. Hence, if b = breadth of belt and t = thickness, then $T_1 = btf$, that is, the maximum pull in the belt ought not to exceed btf pounds. In average practice t = 0.20 for single belting, but varies from 0.16 to 0.25; double belting is supposed to be twice this thickness, but varies in the same way.

Assuming an arc of contact of 180 degrees, a co-efficient of friction equal to 0.27, and the values of t and f as above, it can be shown that the breadth of belt in inches necessary to transmit a given horse power at a known speed, V, may be obtained from

$$b = \frac{800 \times HP.}{V} \qquad (1)$$

for single belt with cemented joint; and

$$b = \frac{1050 \times HP.}{V} \qquad (2)$$

for single belt with laced joint.

Calling this co-efficient C we obtain the general formula:

$$b = \frac{C \times HP.}{V} \quad (3)$$

in which C has the following values:

TABLE I.

$$\text{Values of } C = \frac{Vb}{HP.}$$

	Cement Joint.	Laced Joint.
Single Belt	800.	1050
Double Belt	$\frac{800}{2}$	$\frac{1050}{2}$
Triple Belt	$\frac{800}{3}$	$\frac{1050}{3}$

Rules for Double and Triple Belts.

For belts of double and triple thickness, unless large pulleys are used, there is a loss both in bending the belt around the pulley, and in an actual reduction of the effective arc of contact. When the diameter of pulley is less than 100 times the thickness of belt some allowance should be made for this loss. (Reuleaux.)

The amount of increase should be in some ratio, probably inversely proportional to the diameter of the smaller pulley, and to the square of the belt thickness. It is often customary to use $1 \div 0.7$ as a multiplier for all double belts irrespective of pulley diameter, but while this may be satisfactory in every case, it is hardly necessary except for pulleys less than eight or nine inches in diameter.

For pulleys about 12 inches diameter $1 \div 0.8$ may be used; and for those approximating 20 inches, $1 \div 0.9$.

In the same way for triple belts running over pulleys about 18 inches diameter, the width of belt should be increased in the ratio $1 \div 0.7$; for pulleys about 26 inches diameter $1 \div 0.8$ may be used; and for those about 42 inches diameter, $1 \div 0.9$.

Calling this multiplier, C', we have the values given in the following table:

TABLE II.

Values of C' = multiplier when double or triple belts are used with small pulleys.

For double belts when diam. of pulley =	For triple belts when diam. of pulley =	Value of co-efficient. C'
8 inches	18 inches	$1 \div 0.7 = 1.40$
12 "	26 "	$1 \div 0.8 = 1.25$
20 "	42 "	$1 \div 0.9 = 1.10$

Effect of the Arc of Contact.

Since Table I. was determined with an assumed arc of contact of 180 degrees, the value of the constant C should be multiplied by another co-efficient when the least arc of contact on either pulley differs materially from 180 degrees. This co-efficient, C'' has been determined for different angles of contact, from 120 degrees to 240 degrees, and is given in the following table:

TABLE III.

Values of C'' for different arcs of contact.

Value of Angle— In degrees.	In circ. measure	Value of coefficient C''.
120°	2.09	1.33
130	2.26	1.27
140	2.45	1.21
150	2.62	1.15
160	2.79	1.10
170	2.96	1.05
180	3.14	1.00
190	3.32	.95
200	3.49	.91
210	3.66	.87
220	3.84	.83
230	4.02	.79
240	4.18	.75

If the belt can be arranged so that the tight side will be on the bottom of the pulleys the arc of contact will be increased, since the slack side will thus embrace a greater portion of the upper surface of the pulleys, whereas if the slack side be on the bottom the arc of contact will be decreased when the belt is in operation.

At high speeds this is not as noticeable, since the centrifugal force tends to lift the belt away from the surface of the pulley. In all cases, however, it is desirable to so arrange it where practicable.

Effect of Centrifugal Force.

Thus far we have neglected the effect of the centrifugal force, F°, set up in the belt, but for speeds over 2,500 feet per minute its influence is appreciable and should be considered. The following table gives values of a multiplier or co-efficient, k_o to be used in taking account of the effect of centrifugal force at different speeds.

TABLE IV.

Values of co-efficient k_o = multiplier when centrifugal force is considered.

Velocity in feet per minute.	Cemented Belt.	Laced Belt.
2,500	1.06	1.10
3,000	1.10	1.14
3,500	1.14	1.20
4,000	1.19	1.27
4,500	1.26	1.37
5,000	1.34	1.50
5,500	1.44	1.65
6,000	1.58	1.87

Multiplying these various co-efficients together, we obtain the width of belt:

$$b = C\,C'\,C''\,k_o \times HP./V \quad (5)$$

From this formula and the preceding tables a satisfactory width of belt can be obtained for any given case if the data are known.

Illustrative Examples.

For example, let it be required to find width of double belt to transmit 175 horse power at 3,180 feet per minute, pulleys 90 and 60 inches diameter. As cemented joints should be used we shall have:

$$C = \frac{800}{2} \quad \text{(Table I)}$$

With the diameters of pulleys used it is probable that C'' = 1.05 (Table III.), and k = 1.11 (Table IV.), hence

$$b = \frac{C\,C''\,k_o\ HP.}{V} = \frac{800}{2} \times \frac{1.05 \times 1.11 \times 1.75}{3180} = 24''$$

a belt 19½ inches in width was used.

An electric motor, running at 1,600 revolutions per minute, transmits an average load of 12 horse power through a single belt cement spliced; motor pulley, 7½ inches diameter; driven pulley, 48 inches diameter. Under these conditions the arc of contact is about 140 degrees, and V = 2,950; hence

$$b = \frac{\overset{C}{800} \times \overset{C'}{1.4} \times \overset{C''}{1.2} \times \overset{k_o}{1.10} \times \overset{HP.}{12}}{2950} = 6''.$$

A single belt, with cemented splice, six inches wide is in use. This belt occasionally transmits 15 horse power for several hours at a time.

The horse power which can be transmitted by a belt of given width may be obtained in the same way by transposition of formulae; that is,

$$HP. = \frac{Vb}{C\,C'\,C''\,k_o}$$

For example, suppose it be required to determine how much average horse power a 20-inch double belt (cemented joint) will transmit when running at a velocity of 3,800 feet per minute, considering durability of belt; the smaller pulley is 16 inches diameter and the arc of contact about 155 degrees. Here C = 800 ÷ 2; C' = 1.1; C'' = 1.12 and k_o = 1.17, hence

$$\frac{3800 \times 20}{400 \times 1.1 \times 1.12 \times 1.17} = 135 \text{ HP.}$$

* * *

"Locomotive Engineering" describes a tool-room rack that is in use at the Brainard shops of the Northern Pacific. It is made up of a series of octagonal-shaped tables, one above the other, but each revolving on ball bearings. There are twenty one-inch balls under each rack, which slants to a pitch of about 30 degrees. The slightest touch is sufficient to move any of them. For its purpose it is not easy to conjure up anything that will save more time or steps.

Entered at the Post-Office in New York City as Second-class Mail Matter.

MACHINERY

A practical journal for Machinists and Engineers, and for all who are interested in Machinery.

PUBLISHED MONTHLY BY

THE INDUSTRIAL PRESS,

9-15 MURRAY STREET, NEW YORK CITY.

ONE DOLLAR A YEAR, POSTAGE PREPAID, TEN CENTS A COPY.
FOREIGN SUBSCRIPTIONS ONE DOLLAR AND FIFTY CENTS A YEAR.

Lester G. French, Editor.
Fred E. Rogers, Associate Editor.

The receipt of a subscription is acknowledged by sending the current issue. Remittances should be made to THE INDUSTRIAL PRESS, and not to the Editors. Money enclosed in letters is at the risk of the sender. Changes of address must reach us by the 15th to take effect on the following month; give old address as well as new. Domestic trade is supplied by the American News Company or its branches.

We solicit communications from practical men on subjects pertaining to machinery, for which the necessary illustrations will be made at our expense. All copy must reach us by the 5th of the month preceding publication.

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APRIL, 1899.

The regular edition of MACHINERY for April is 15,250 copies, which is the largest circulation in its field, and to those interested we shall be glad to furnish proof of this statement. AMERICAN MACHINERY is the title of the foreign edition of this journal, which is printed on thin paper and comprises all the reading and advertising matter in the domestic edition.

* * *

INACCURATE SCREW THREADS.

It is an indisputable fact that in the matter of producing screw threads of accurate pitch the art of machine construction is not so far advanced as in its other branches. The manufacturers of micrometers use short screws for all sizes of the instrument, and depend upon distance pieces or other devices for obtaining the larger adjustments, because of the difficulty of producing long, accurate screws. In general machine work we have cylindrically ground journals and bearings, scraped surfaces and accurately cut gears, both spur and bevel; but the threaded pieces are not, in most cases, examples of good workmanship.

This is not only true of rough work, like machine bolts and screws, but is too often true of the better class of work, where accuracy is to be expected. We believe it to be an exceptional case when a screw can be cut in one lathe which will properly fit a nut threaded in another lathe made by another builder, and oftentimes when made by the same builder. The discrepancy may not be very noticeable with a nut of ordinary length, but if the nut be several diameters long the screw will usually have to be turned below size in order to pass through the nut.

The difficulty is aggravated, moreover, when, as is usually the case, the pieces with internal threads are either tapped or sized with a tap after having been roughed out in the lathe. Of all the tools in the shop the tap is, perhaps, the most inaccurate. This is not laying up accusations against tap manufacturers, for the skill of many of the best mechanics has been taxed to its utmost to produce taps of true pitch; but as far as we know, no method for correcting or avoiding the spring due to hardening has been made a commercial success.

We are led to call attention to this matter of screw threads through the experience of a company building large steam engines. They have been having a great deal of trouble in threading the various parts of their engines which screw together, such as piston-rods and cross-heads, adjusting wedges and bolts, eccentric-rods and straps, etc., so that the pitches would come alike and the parts go together.

Their troubles are but common experiences in every shop. The screw-thread problem is ever present, and but little can be offered in the way of its solution. The first step to be taken is clearly to have the lead-screws of the lathes made correct. It is commercially practicable to cut lead-screws for engine lathes of small and medium size whose pitch is accurate, within .002 inch per foot. We say that it is commercially practicable to do this because we know that certain builders are turning out lead screws of this character, as determined by tests in shops where the lathes have been sold. There are doubtless other builders doing equally as good work and we doubt if any reputable builder would refuse an order at regular rates for a lathe of this size where a maximum limit of .002 inch per foot was specified for the lead-screw. The admission that the product was not up to this standard would be a damaging one.

As far as taps are concerned, the trouble is not so easily eliminated. Small, solid taps answer very well for tapping out short holes, and the remedy for their inaccuracy is obviously to avoid long internal threads when designing the machine.

Good results can be secured with large taps by making the bodies of soft steel and inserting tool-steel cutters. This is what is done in the most successful dies and by carefully hardening the chasers or cutters equally as good results should be secured with either taps or dies. One shop where we have called has a large set of adjustable taps made on the principle of adjustable reamers, having five or six threaded blades fastened to one body, and very good results were claimed for them. They have all the advantages of the adjustable reamer and the added advantage that the small cutters could be hardened without much distortion. Still another plan for taps is to drill holes at right angles to the axis of the body of the tap, along the path of the spiral, and to insert round cutters, which are then shaped by turning, with the tap body between the lathe centers.

One important point that is generally overlooked in tapping long holes, or in tapping any hole with a large tap, is that if the pitch is to be preserved the tap must be led, either by a screw or by threads following in the part already tapped out. Many make the mistake, however, of trying to lead a tap of inaccurate pitch, with the result that the thread becomes torn and distorted. One of the best dies on the market is constructed so as to first secure chasers of correct pitch, and, secondly so that the die will be led by blunt chaser threads following in the wake of the cutting edges. If this combination could be secured in a tap, we believe that good results would follow.

* * *

We are glad to present in this issue some of the features of the tallest office building in the world, recently completed in this city. The modern tall building is a structure, the designing and erection of which comes strictly under the heading of engineering. The structure itself, from the foundations up, is a massive piece of structural steel construction, in which is installed in some cases as much machinery as would be found in a large manufacturing plant. The whole is encased in a stone and brick covering for protection against the elements, and for architectural effect, but supported entirely by the frame work. The subject of the tall building and its machinery is one of the greatest interest from an engineering standpoint. In this connection, it may not be out of place to call attention to the excellent photograph of the building shown on the first page. The building is so tall and the available space in front is so small that several photographers have pronounced it impossible to get a rectilinear photograph without distortion. The view shown is the first successful one that has been made, and was taken especially for MACHINERY by Reichert & Henius, New York. Another interesting point about the photograph is that it was taken when the street was crowded, but, as the exposure was timed the movement of the people and the vehicles caused them to become obliterated.

* * *

See that the drip cups on the shaft hangers are occasionally emptied. A few drops of dirty oil down the back of the neck are often the cause of much needless profanity.

SHOP TALKS WITH YOUNG MECHANICS—13.

GAUGES AND OTHER TOOLS.

W. H. Van Dervoort.

There is nothing more confusing to the young mechanic than the use of the several systems of gauges used in designating the sizes of wire, machine screws, drills and plate thicknesses. Unfortunately, most of these dimensions differ from each other for corresponding numbers by comparatively small amounts, yet an amount sufficient to cause error if the one is mistaken for the other. The following table gives for comparison values for only a few numbers under each of the several gauges in most common use:

No. of Gauge.	American or Brown & Sharpe Gauge.	English, Birmingham or Stubs' Wire Gauge.	Imperial Wire Gauge.	Stubs' Steel Wire Gauge.	Stubs' Drill Gauge.	United States Standard Plate Gauge.	Steel Music Wire Gauge.	Standard Machine Screw Gauge
1	.2893	.3	.3	.227	.228	.28125	.0156	.071
5	.1819	.22	.212	.204	.2055	.21875	.0202	.124
10	.10189	.134	.128	.191	.1935	.14062	.027	.189
15	.05707	.072	.072	.178	.18	.07031	.0345	.255
20	.03196	.035	.035	.161	.161	.0375	.0434	.321

The dimensions are purely arbitrary. The American, or Brown & Sharpe gauge, was brought out in the production of a gauge to overcome the irregularities in spacing of the Birmingham wire gauge. In this gauge the dimensions increase by a regular geometrical progression. The largest dimension is No. 0000, which equals .46 of an inch. The next smaller number, 000, is obtained by multiplying .46 by the constant .890522. This product again multiplied by the same constant, gives the next smaller number, and in like manner each number is the product of the preceding number and this constant. A comparison of the English and American gauges is best shown in Fig. 225, where the peculiar irregularities of the former are plainly shown.

The imperial wire gauge differs but little from the English. It was adopted by the English Board of Trade in 1884 as a substitute for the Birmingham gauge. The Stubs' steel wire gauge differs materially from those above cited. Its range carries it from No. 1 to No. 80, variations in dimensions being indicated to the nearest one thousandths of an inch. The difference between consecutive numbers differ in dimensions from one to at most only a few thousandths, the eighty numbers carrying it through but .214 of an inch.

It is extremely unfortunate that a single standard gauge could not be adopted and used to the exclusion of all others, at least in our own country, and thereby avoid all the confusion incident to the promiscuous use of the several so-called standards. Even the manufacturers of brass and copper stock, who have adopted the American standard, do not confine themselves exclusively to this gauge, as much of their product is still gauged by the English system.

The Stubs' drill gauge varies from the Stubs' steel wire gauge by from 0 to 3 thousandths of an inch over size, which is simply the average oversize determined by a great number of measurements of wire drawn through Stubs' wire gauge dies, and might be compared with a maximum limit gauge, as the dies, when worn to such an extent as to produce wire over the sizes indicated by the drill gauge are replaced by new ones.

It will be noted that as the designating number of the gauge increases the dimensions decrease for all except the steel music wire and machine screw gauges, which increase in diameter as the number increases. This also tends much towards confusion, and may be looked upon as another anomaly of the present gauge systems.

The gauges, or tools for indicating the gauge of wire or plates, are of two forms, the angular and the notch. The angular gauge is shown in Fig. 226. With this tool the measurement is made

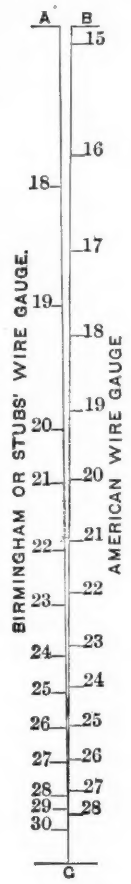


FIG 225.

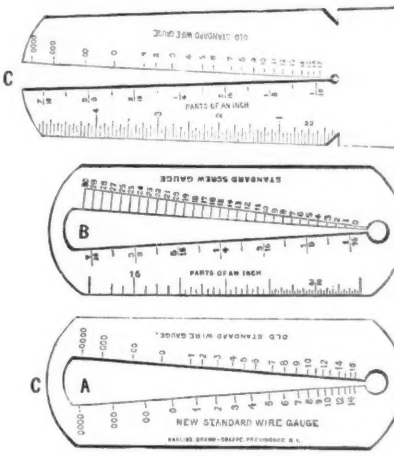


FIG. 226.

FIG. 227.

by passing the screw, wire or plate into the angular opening until it touches both sides; the reading opposite the point of contact giving the gauge of the material. When used for gauging plates, this gauge should be made with open end, as shown at C. On the one side is graduated the English and American standards, and on the other the machine screw gauge and parts of an inch. The notch gauge is shown in Fig. 227. In using this tool the article measured should just pass through one of the slots, the number opposite indicating the gauge. These gauges may be had with the decimal equivalent of each size stamped on the back of the tool, which will frequently be found to be a great convenience. Another form of notch gauge is shown in Fig. 228. This is called a rolling mill gauge, and is used for gauging sheet metals. They may be had either in the English or United States standard plate gauges.

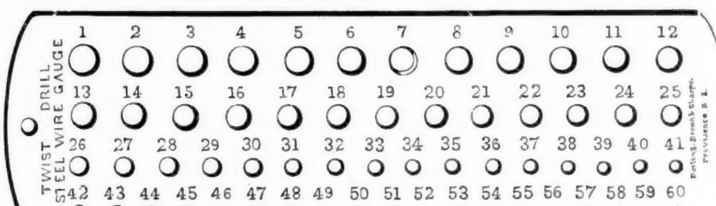


FIG. 229.

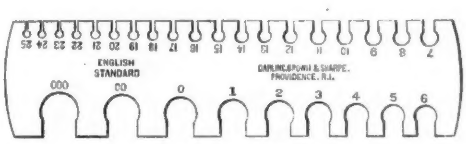


FIG. 228.

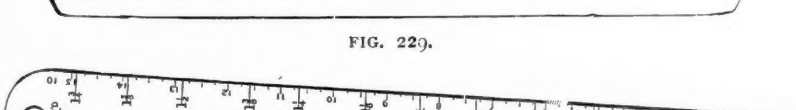


FIG. 230.

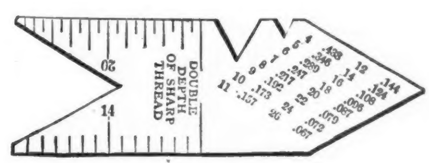


FIG. 231.

As there is considerable wear upon gauges of this class, and

especially those of the notch pattern, it is important that they should be made of good steel, carefully hardened and tempered. In order that tools of this character can be put upon the market at a reasonably low price, high accuracy requirements in their manufacture cannot be attempted. For all ordinary gauging they will be found sufficiently accurate and when greater exactness is demanded the micrometer caliper had best be used, the decimal equivalent of the gauge required being taken from a table or the back of the gauge.

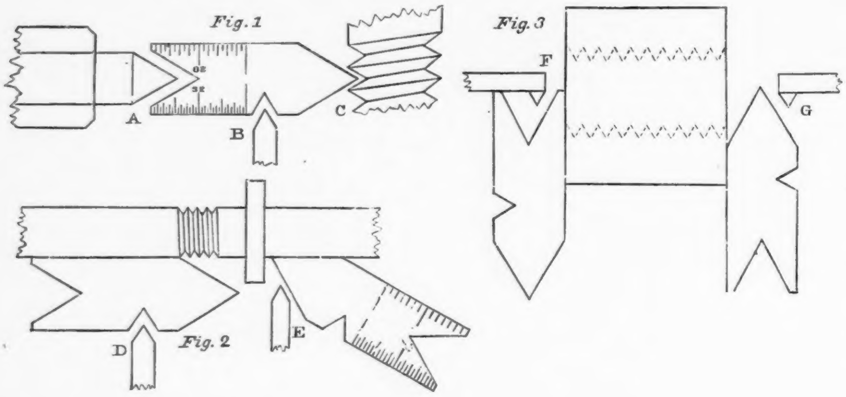


FIG. 232.

In Fig. 229 is shown a drill gauge, Stubs' drill sizes, from No. 1 to No. 60. A smaller size, with holes from No. 61 to 80, is made. It may also be had with holes from 1-16 to ½ inch, varying by 64ths, the latter being known as the "jobbers'" drill gauge. As it is not practical to attempt to stamp the sizes on very small drills, these gauges are quite necessary.

The nut and washer gauge shown in Fig. 230 is so graduated as to show readily the diameter of holes within its capacity. A very convenient feature in this gauge is the dimensions giving sizes of holes to tap United States standard threads. A gauge of this kind will be found excellent for measuring small holes or narrow slots, which are too small to be calipered, and frequently in inaccessible places where they cannot be scaled.

In Fig. 231 is shown a center gauge, which, as its name indi-

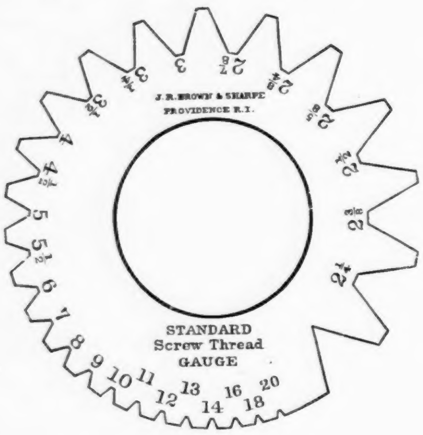


FIG. 233.

cates, is used for gauging lathe and other machine centers, in turning or grinding them. It may also be used as a gauge for grinding and setting threading tools, a number of its uses in this connection being clearly shown in Fig. 232. The angles are 60 degrees, and the table of equivalents shown is to determine the proper diameter of hole for tapping full V threads. In this table the numbers in the first and third columns are threads per inch, and those in the second and fourth columns the double depth of V threads of the corresponding pitches. Thus the tap drill for a ¾ tap, 10 threads per inch, would be .75 — .173 = .577 =37-64 inch.

The screw thread gauge is shown in Fig. 233. The one 60 degree V notch will gauge all V threads; the others, however, are flattened at the root by the amount the top of the thread is cut off in the various pitches of the United States standard thread. These gauges are for use in grinding threading tools, and may be had for worm, Whitworth and "acme standard" threads.

The screw pitch gauge, Fig. 234, has a large number of thin blades, which fold up into a suitable handle. Each blade has two or more teeth, of the pitch corresponding to that stamped on the blade. By direct comparison with a screw thread the exact pitch may be determined without the danger of errors that arises when the threads are counted over the edge of a rule. The decimal following the pitch number on each blade is the double depth of the thread. The thickness gauge, shown in Fig. 235, consists of a number of thin steel leaves, varying by thousandths in thickness. The leaves may be used singly or together, thus making any thickness desired within the limits of the tool. The value of a tool of this kind can only be learned by having one in the vest pocket, where its convenience to hand will find many chances for its use.

The depth gauge, an example of which is shown in Fig. 236, is used for measuring the depth of holes or recesses. The blade should be narrow, and for general work graduated on one edge to 64ths and on the other to 100ths. The beam or head, when six inches long, will meet most general requirements. This tool will be most highly appreciated by the man who has often attempted to measure carefully the exact depth of a recess by holding two slippery steel rules at right angles to each other between his thumb and fingers, and attempted to read the one over the edge of the other. For very accurate depth measurements these gauges are made in a micrometer pattern, which is graduated to read directly to thousands.

While on the subject of gauges we should not overlook the simple yet useful one shown in Fig. 237. The scratch gauge con-

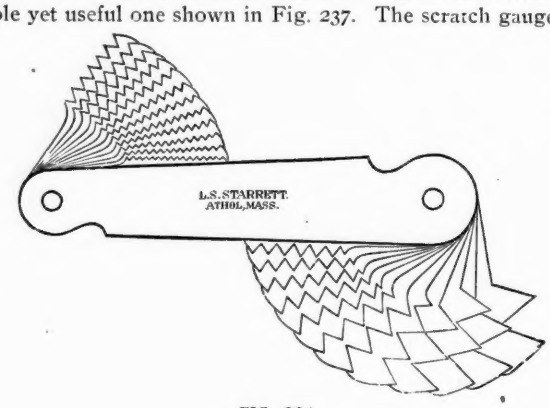


FIG. 234

sists of an arm, preferably graduated, carrying a sliding guide, which can be clamped at any position on the arm; and provided with a short, hard and sharp scriber at the end. This tool is used in ruling lines parallel with the edge of work.

In Fig. 238 is shown one of the many types of surface gauges.

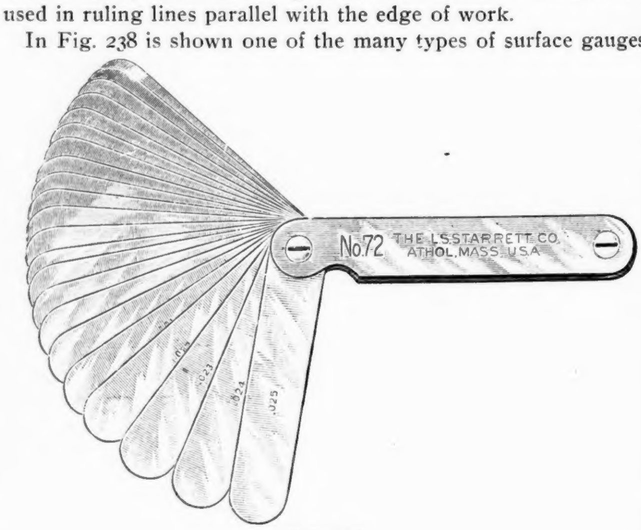


FIG. 235.

Although this tool comes last in the long list of useful gauges above illustrated, it is by no means the least important, as the machinist undoubtedly uses it more in general machine shop operations than all the others combined. The principal use of this tool is in determining the parallelism of one plane with an-

other, usually the surface of the work with the machine table, housing, cross-rail or other reference planes. In testing, erecting and setting up work on machine tools the surface gauge is indispensible.

The simpler the gauge is made the better it is, as numerous tricks and devices on a tool of this kind usually complicate and decrease rather than increase its value. In setting the point of the needle to a line or point the clamping screw should be so adjusted that the needle moves smoothly yet with considerable friction. The power applied to the needle to turn it should be on the opposite side of the fulcrum from the work, as in that case the spring of the needle is outside of the fulcrum and the point stays where placed. If the force is applied inside the fulcrum the spring will make it difficult to set the point as desired. When, as with the gauge shown, a screw adjustment is provided, the exact setting is made with this adjustment.

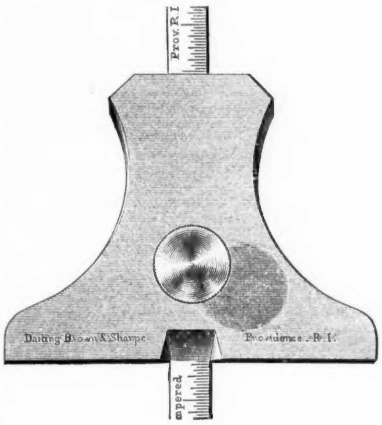


FIG. 236.

In Fig. 239 is shown an attachment to an extension divider. This ball point, taking the place of one of the divider legs. The balls which are of different sizes are rather more than half spheres, hardened and ground. The great value in this point lies in its adaptability to scribing circular arcs around the center of a bore. It avoids the necessity of plugging and centering the bore, and will be found a reliable and useful tool.

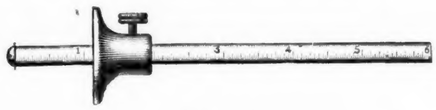


FIG. 237.

A pair of tramel points are shown in Fig. 240. The beam is usually made of wood. This tool takes the place of the divider in scribing arcs of long radius. Inside or outside caliper legs may be inserted, in place of the points in the heads, thus making calipers of wide range.

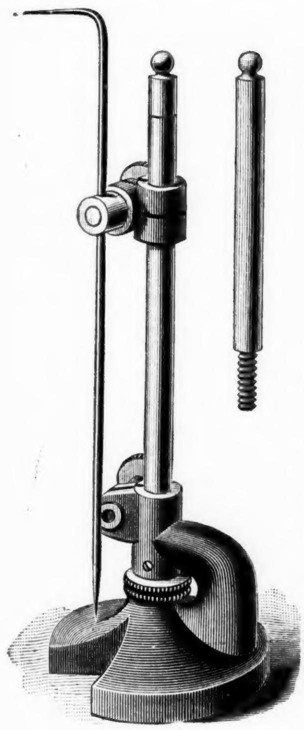


FIG. 238.

FIG. 239.

A test indicator is a tool used in determining small variations from the time rotation of a cylindrical surface and irregularities or inaccuracies in its cylindrical truth. It can also be used in determining the inaccuracies of a plane surface, and small amounts of end or lateral motion, as for example, the end motion of a spindle or the deflection, give or wink between gibbed surfaces. These tools are of two types; those which simply indicate, and those which give a reading that shows the exact amount of the error or untruth. In Fig. 241 is shown an instrument of the latter class. The adjustments of this tool are quite evident from the figure. The long pointer, the one end of which moves over a graduated arc with readings to one one-thousandth of an inch, as fulcrumed, bears the hardened point, which comes in contact with the surface to be tested. The reading is magnified by the long pointer, and the zero of the scale is at the center of the arc, which reads ten thousandths of an inch each side of this point. A light spring, secured to the pointer, and held between the adjusting screws near the pivot, provides for the convenient adjustment of the pointer to the zero reading, no matter what the position of the arm.

Instruments of this character must be carefully made, and are of great value in the erection and testing of accurate machinery. When, however, only an indication of untruth is required, as in the chucking, centering or setting up of work, a much cheaper tool serves the purpose, as for example, the one shown in Fig. 242. In this tool the pointed pointer is held in a universal socket, which is carried on the end of a bar of suitable form to clamp in the tool post of the lathe. If the point is brought against the rotating work, the amount of motion at the outer end of the pointer indicates the extent to which the work is out and the way in which to move it in truing. It is a superior method of truing nice work, which will not injure its surface, as is so apt to be the case when turned to the point of a lathe tool.

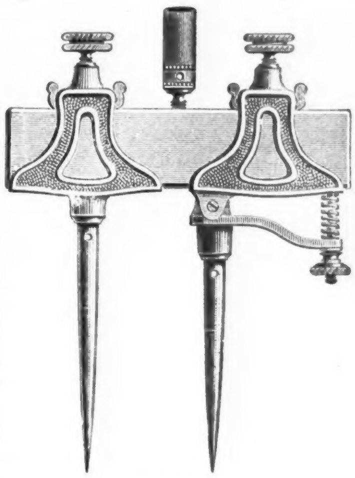


FIG. 240.

One of the neat applications of this tool is in the centering of a piece of work, in the chuck or against the face plate, to a point. The sharp end of the indicator pointer is set in the point to be centered, and the work revolved. This causes the outer end of the pointer to describe a circle, the diameter of which determines how much the point is out of center. By properly setting the instrument this circle will be described around the tail center, and when the work is exactly centered the pointer remains stationary in front of the tail center.

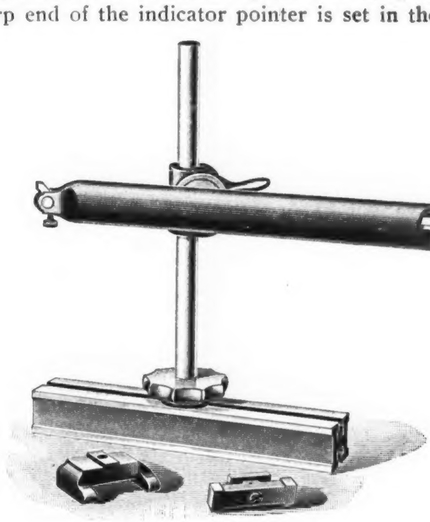


FIG. 241.

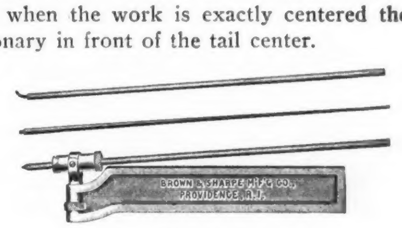


FIG. 242.

The above serve to illustrate a few of the many applications of an exceedingly satisfactory, yet not extensively used, class of test tools.

* * *

Scraping wrought iron or steel to a surface plate is a tedious operation when done dry, as the best scrapers will tear and scratch, but if the tool be dipped in turpentine once in a while the result will be a smooth surface, and the time required will be greatly reduced.

PATTERN SHOP NOTES.

W. H. SARGENT.

STORING SUPPLIES.

What a tinker a pattern maker is! Iron and brass, pine and mahogany, leather, wire, wax and putty are his materials. His tool-box contains more "ingrediencies" than the witches' broth in Macbeth. A hundred sizes of brass and iron screws, half as many of wire brads, gimp tacks, carpet tacks, staples, screw eyes, copper rivets, iron rivets, burrs and washers, all these and more, so arranged as to be instantly found when wanted, however does he do it? If he is smart he procures a lot of large-mouthed "homeopathy" vials with corks and fills them up with his specimens. He can easily arrange a case for them, or he can pack away a large number in a drawer so that any one can be readily found. For anything within the limits of its size such a bottle has certain advantages over a box or wrapper; the articles are always visible without opening the package or spilling the contents, and are easily shaken out into the hand just as one would shake out a pill to take after eating.

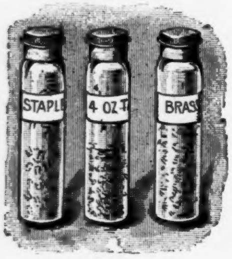


FIG. 1.

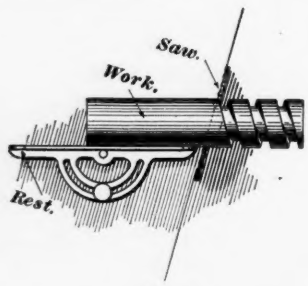


FIG. 3.

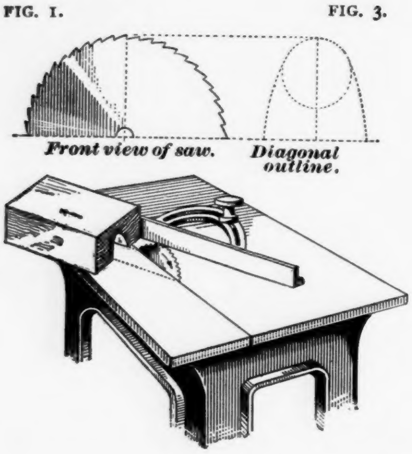


FIG. 2

NEW TRICKS WITH AN OLD SAW.

Sometimes in making a round core-box it is convenient to have some way of removing the stock instead of grubbing it out with "hollows and rounds." Our old friend the circular saw grins until he shows his teeth and offers his services to aid the work, and the accompanying sketch shows how it can be done. By setting the splitting gage or "fence" on an angle the work will be fed, not directly at the edge of the saw but more or less on the "bias" as shown in Fig. 2. The angle of the fence may be varied but must be in the same direction as shown in the sketch, so that the work will crowd against it and not away from it. As shown by the diagram, the saw when approached at an angle takes on the form of an ellipse, no part of which is a true circle, so that the saw should be large as compared with the diameter of the groove which it is expected to cut. If the saw is stiff enough not to spring, and the work is fed to it slowly, not taking too deep a cut at a time, this method will be found fairly successful, and in the words of the old saw "you never saw a saw saw as this saw saws."

"TWISTING AND TURNING DONE HERE."

says an old sign over a cabinet maker's shop in a college town. When the boys tacked up this sign over a lawyer's office the people allowed that the wording was not inappropriate. Neither would the sign be out of place over a circular saw, for if a shop is not equipped with a lathe for turning twisted work, a saw as arranged in Fig. 3, will be of great assistance. The work to be "twisted" is first turned out in the lathe, the cross-cut "fence" on the saw set to the necessary angle and located so that the center of the work will come over the center of the saw. When the saw is set to cut to the proper depth, the work, if slowly rotated over it, will be drawn along across the saw in much the same way as a die follows the thread which it is cutting. By going over the work several times a good square thread can be made which can be worked into shape with a knife. Useful for making a pattern of a coarse screw for any purpose, as a winding drum for a hoisting engine. Both of these tricks are illegitimate uses of the saw and are therefore attended with some little danger and unusual care should be employed.

* * *

DEGREE OF ACCURACY REQUIRED.

The following taken from an English paper is worthy of consideration, as very frequently the parallel in one way or another is seen here:

"Considerable time and expense might be saved in the fitting up of special tools if the home shop were informed as to what degree of accuracy was required on that particular piece. As a case in point, a manufacturer in this district placed an order with an American firm for an automatic machine, to be fitted up for making steel bushes. The representative who received the order, not going into the question as to what quality of work was required, sent the model across to the firm, who proceeded to fit up the job in accordance with its usual custom on that class of work. The machine was arranged to accurately turn the outside of the stock, and finish the hole in the center of piece to size, by the use of reamers, and produced work exactly to model. The machine rigged up with expensive tools and full set of extra tools as well, arrived duly and was put in operation producing an accurate job. A few days later the American representative calling there noticed that the outside turning tools and the reamers had been removed from the machine, and it was simply performing the operation of drilling and cutting off the bushes. Upon his inquiries as to the cause of the removal of the tools especially made for producing accuracy in the work, the manager of the concern pointed to a row of English hand machines operated by girls, drilling and cutting off the same style of bush, and told him that if he should send work as produced on the automatic, dead to size, along with the work from the other machines to his customers, they would reject all the bushes made on the old machines. In other words, the machine as arranged did too good work for him, and the cost of getting out the special tools might as well have been saved had the representative been informed as to what sort of a job was wanted."

* * *

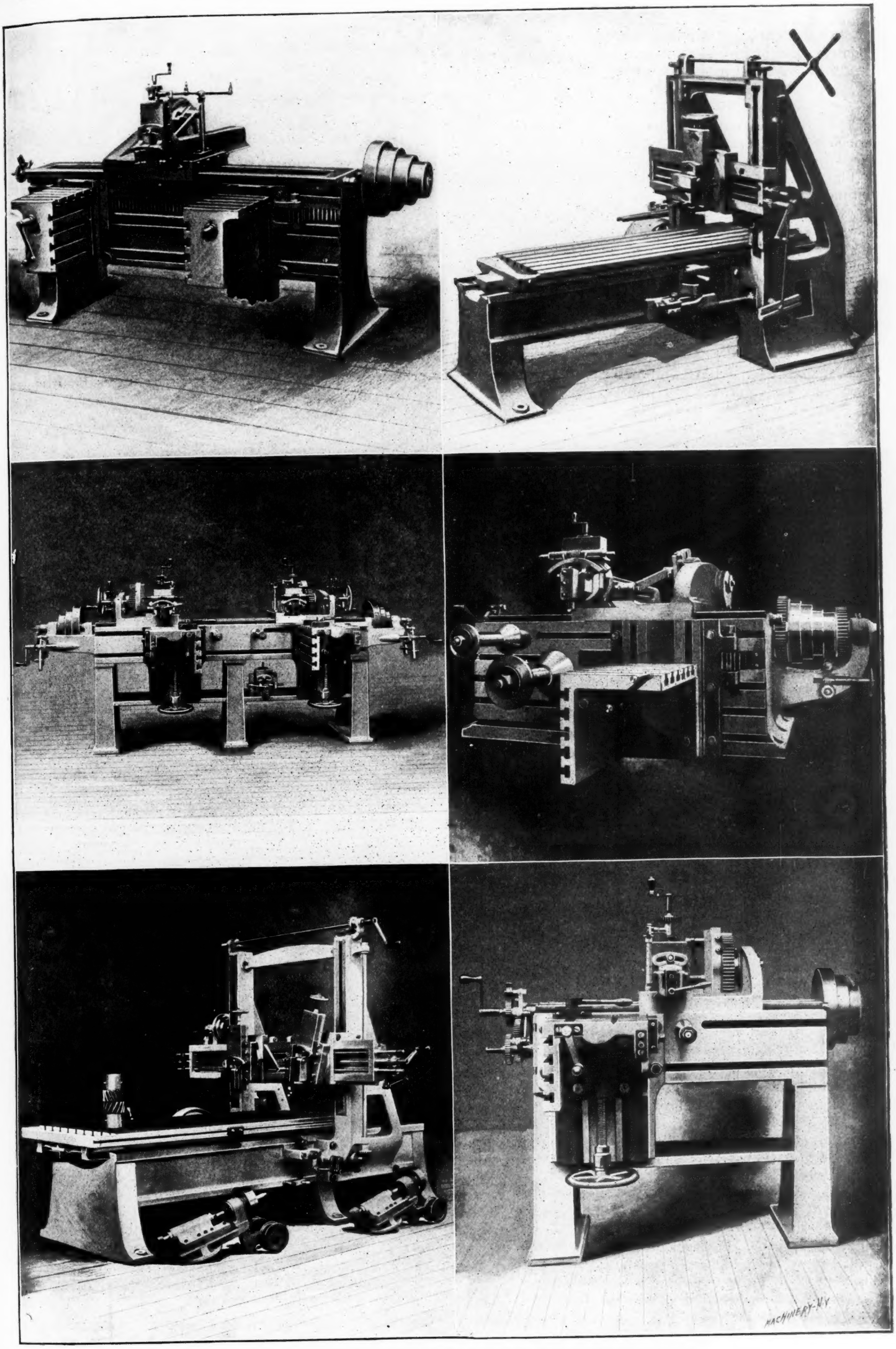
TYPICAL EUROPEAN MACHINE TOOLS—5.

SHAPERS AND PLANERS.

In this issue on a full-page illustration are shown some European shapers and planers. Of shapers a large variety is to be found. This machine is being built in even very large sizes, both with one and two heads, the later kind for use on connecting rods and similar work. There is, however, no question that the modern plain milling machine encroaches greatly on the usefulness of the shaper. This tendency is being felt abroad, where the growth of the plain milling machine for general and special work has been slower then with us here. The illustration shows the conventional crank-construction. It would almost seem that the designers in neglecting to improve this type realize that their efforts directed in other lines promise larger returns. A peculiarity noticed in the cuts is the horizontal arbor fixed on the side of the bed to allow for shaping grooves, or flats around the circumference of a piece. While in theory, quite proper, in practice this arrangement is not much used. In regard to planers, designers have been stimulated by the exhibit of the Sellers Company at the last Paris Exposition.

The machines of that exhibit evoked a great deal of favorable comment for the rapidity with which they performed the work. But the tendency of European designers to apply milling and drilling attachments to their planers prevented to a great extent the use of fast platen speeds. True to their preference for vertical milling machines the designers employed in these attachments vertical spindles, carrying end-mills. To make a universal machine out of a planer is not favored by our machine-tool builders, who prefer to keep the different types distinct, and with very good reasons. As to the method of driving the platen the well known different means are found represented, such as, the plain rack, the herring-bone rack, and the Sellers worm rack drive.

C. C. S.



EUROPEAN MACHINE TOOLS—SHAPERS AND PLANERS.

SHOP KINKS.

A DEPARTMENT OF PRACTICAL IDEAS FOR THE SHOP.
Contributions of kinks, devices and methods of doing work are solicited for this column. Write on one side of the paper only and send sketches when necessary.

SOME USEFUL KINKS.

The following simple and useful ways of doing work are contributed by E. D. Haddix of Burlington, Iowa, and while well-known to many they probably will be new to many others.

Fig. 1, is a small cast iron chuck threaded at end B to screw on the lathe spindle and bored out at C quite large for the two jaws shown in Fig. 2. The set-screw D, screws down on the jaws holding them and also the work in the hole H. The jaws are bored out in different sizes and are useful for holding round steel rods for studs, etc. Those shown in Fig. 2, are bored for two sizes.

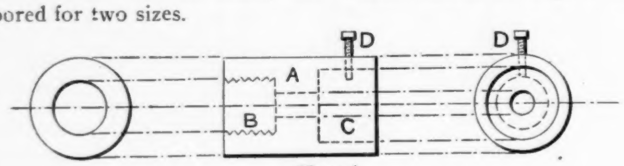


FIG. 1

FIG. 2

FIG. 3

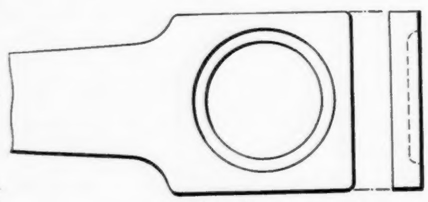


FIG. 4

Fig. 3, is a flat piece of steel having a cupped cavity bored on the lathe so as to be true and is used for a die in forging small washers. The washer is heated to a forging heat and then placed under the steam hammer and hammered into the die. It makes a fair job as the washers need only a little grinding and polishing on the buffing wheel after they are drilled. Fig. 4 is a top view and Fig. 3, a section.

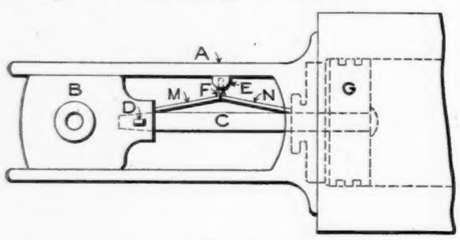


FIG. 5.

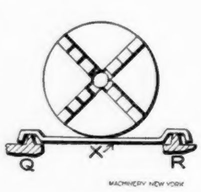


FIG. 6.

Fig. 5, shows how we removed a tight fitting piston-rod from the crosshead without much trouble. The crosshead B was pulled clear back till the piston G struck the back head. The jack E was placed in position between the frame A and the pieces M and N. These were arranged so that the effect of a toggle joint was obtained, one end of M resting against the crosshead, and the end of N being against the cylinder head, the jack being placed as shown in Fig. 5.

Fig. 6, shows a way to put on and remove heavy lathe chucks. X, is a strip of flat iron ½ in. x 2 in. bent as shown and laid across the ways Q and R of the lathe. It is just the right height to slip under the chuck when on the spindle and with it a lathesman alone can easily put on or remove a very heavy chuck.

PLANING A KEYWAY.

"G. G." sends a method for holding a shaft longer than the planer platen when a keyway is to be planed out. A glance at Fig. 7, with his description will make his method clear. The shaft A projects beyond the platen and the bolts C C, are provided which are threaded at each end. These pass through the strap D and the pieces F F. The piece D thus takes the thrust

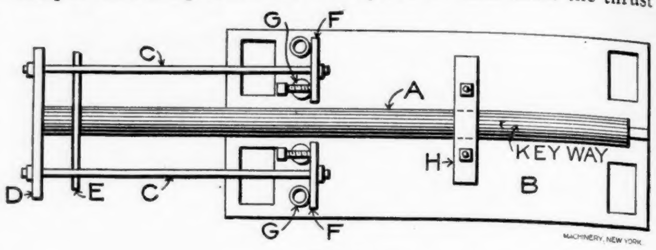


FIG. 7.

of the tool and E is bent over the shaft and under the bolts C C to hold the shaft down. It is also strapped at H to the platen.

A KEYSEATER.

A writer, who signs "Jake" says, that in the January number of the MACHINERY a keyseater was shown from "Uncle John"; so he sends a sketch of one that he thinks will be easier to make. It will not need much explanation as it is very simple in construction as can be seen by reference to Fig. 8. It is used in the shaper, small work being held in the vise and large work

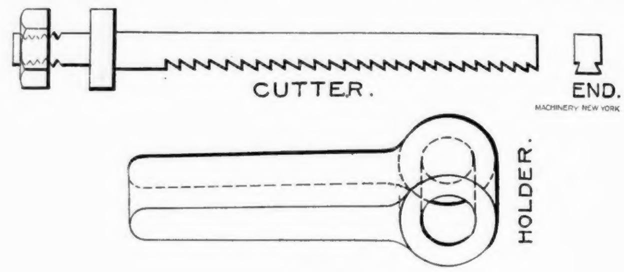


FIG. 8.

being bolted to an angle plate provided. It has proved to be a very useful tool. The angle plate has a slot for the cutter to pass through. By having different widths of blades to fit holder it is no trouble to change from one size to another. The holder should be made as large as will go into the tool post so there will be as little spring as possible. The teeth should have but little rake ahead, as too much gives a tendency to dig into the work.

DEVICE FOR PUTTING ON CLAMPS.

It often is a patience trying job to put on a hose clamp so that a nozzle or union can be held, especially if the clamp is a little small and you want to put in a short bolt.

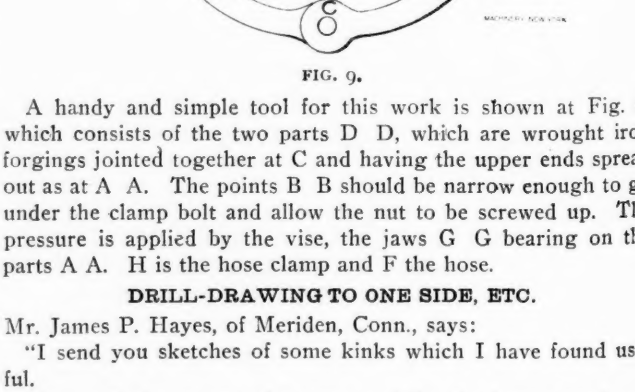


FIG. 9.

A handy and simple tool for this work is shown at Fig. 9, which consists of the two parts D D, which are wrought iron forgings jointed together at C and having the upper ends spread out as at A A. The points B B should be narrow enough to go under the clamp bolt and allow the nut to be screwed up. The pressure is applied by the vise, the jaws G G bearing on the parts A A. H is the hose clamp and F the hose.

DRILL-DRAWING TO ONE SIDE, ETC.

Mr. James P. Hayes, of Meriden, Conn., says:

"I send you sketches of some kinks which I have found useful.

Fig. 10, B shows part of a screw and the check nuts used as a stop for a hand milling machine. This particular miller is used entirely on jobbing work, and consequently the check nuts have

to be adjusted constantly, and are frequently run nearly the whole length of the screw. I replaced this screw (which is a ½ in. 14 thread) by a quadruple threaded screw of 3½ in. pitch shown at B, Fig. 10. An equivalent single threaded screw is shown at C, Fig. 10. This screw saves time and nerves and can be adjusted within .002" which is near enough for most milling machines or drill presses.

I recently made a six spindle countersinking attachment for a drill press which was used for countersinking the holes in some 3½ in. brass discs. When tried the countersinks all drew in to one side of each hole as shown. The disc which was free to turn on a pin in the center hole of the disc was drawn in the direction of the arrow, Fig. 11. The spindles all run left handed.

The only explanation I can give is that when the countersinks are cutting on the sides of the holes which are farthest from the center of the disc, there is a greater leverage than when cutting upon the opposite side of the holes. If this explanation is correct then in drilling, reaming or countersinking a hole in a piece as at 11A with a stop B, the drill would tend to run towards side X. This tendency is probably not new to a great many but it was a great surprise to me and I shall hereafter look out and not get caught again. This tool was corrected by making cutters with tits on them.

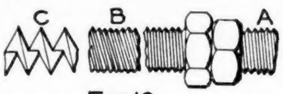


FIG.10

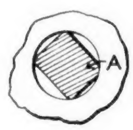


FIG.11 A

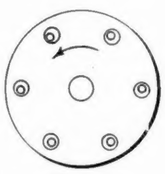


FIG.11

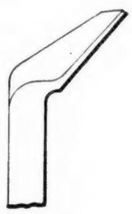


FIG.12

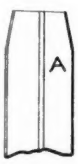


FIG.13

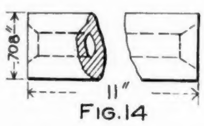


FIG.14

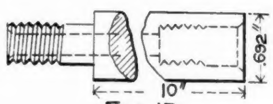


FIG.15

As I reckon it, the leverage in this case is about as 7½ is to 8½. The more holes or the nearer the center of the disc the greater would be the tendency to draw around.

The lathe tool shown in Fig. 12 is not commonly seen in the shop, but it deserves to be on every lathe board. It is a left side tool bent to the right and can be used close up to the dog or chuck.

The square reamer shown in Fig. 13, is for making a very smooth and accurate hole. This reamer has been used on work when nothing else would meet the requirements as well.

In flatting the sides it should be made thinner one way than the other so as to give irregular spacing of the cutting edges. There is a little land left at the corners which after hardening is backed off slightly with an oil stone. The flats A, A, should be very smooth.

Fig. 14 and 15 illustrate a cutter shank which until recently has always been made of tool steel left unhardened as the ends Fig. 15, had to be very true and accurate. The shank fills a hole nicely and goes through ahead of the cutter which enlarges the hole .03", and which fits into the female end.

I now make the shank of common soft steel as in Fig. 14, and then have it carefully hardened in water, keeping them in the pot about five hours. Before hardening the holes are plugged with graphite paint which prevents them from hardening. The shanks do not spring badly and they can be easily straightened before grinding. I grind the hard scale off on the ends, and can then chuck the hole in one end and cut the thread on the other as the hard shell does not extend more than a 32nd of an inch deep. When ground to size, I have a shank with a surface as hard as a file and a soft center. I think spindles, etc., might be made in this way with good results."

TO TURN ECCENTRICS OF UNIFORM THROW.

Out in a Chicago shop the rig shown in Fig. 16, was devised for turning eccentrics of an uniform throw. In describing it Mr. Karl Keely says:

"We had some eccentrics about 12 in. diameter, 2¾ in. throw, to turn and it was necessary that the throw and eccentricity with relation to the keyseat should be exactly the same in each one. This was accomplished with good results in the following simple manner:

All eccentrics were bored and keyseated first; then a flanged casting was turned and keyseated to fit eccentrics. The cast-

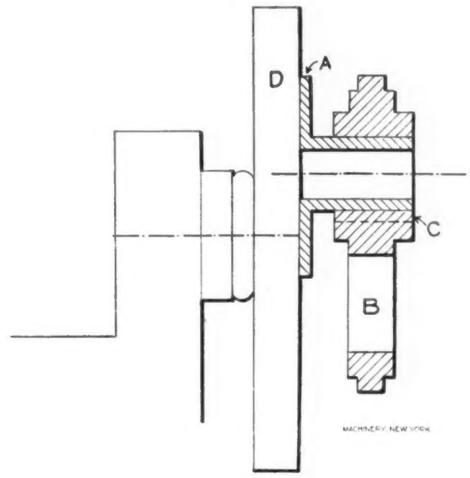


FIG. 16.

ing was fastened securely to the face plate off center enough to correspond to the throw of the eccentric.

The eccentrics were held in place by means of a key, and the casting remaining undisturbed until the job was completed. They came off as nearly alike as could be desired."

DEVICE FOR TURNING BALLS.

The device for turning brass balls shown in Fig. 18, we think will be new to some readers of MACHINERY. It was made for turning five-inch brass balls on a drill press and the cut shows the essential features. The balls which are used as valves for a form of water pump are mounted on a madrel ⅝" in diameter. The mandrel and ball are turned by a ratchet feed worked by a cord attached to an eccentric on the drill spindle.

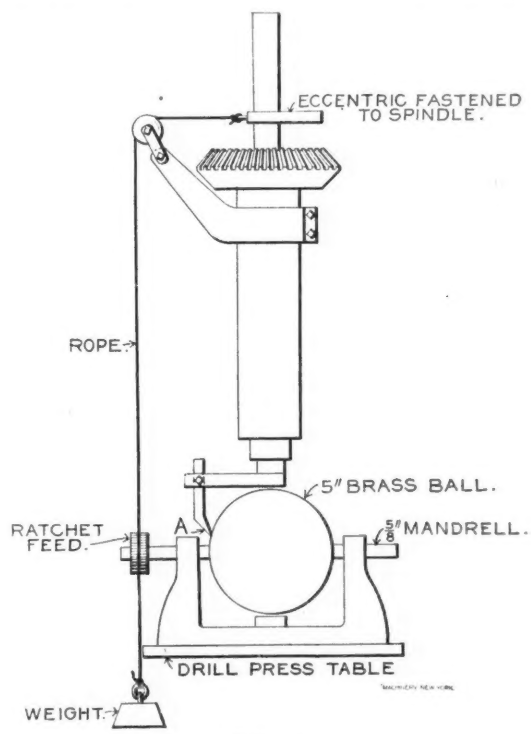


FIG. 17.

The cutter A has no traversing motion but simply sweeps through a circle of unvarying diameter, the rotation of the ball presenting all points except the poles to the cutter. After the balls are turned the mandrel is driven out and the holes plugged and smoothly finished off.

Mr. C. W. Gleason of New York City, sends us the description and sketch of the rig.

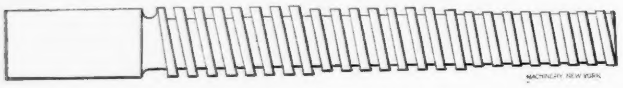
LETTERS UPON PRACTICAL SUBJECTS.

A THREE-STEP TAP—NOVEL PUMP.

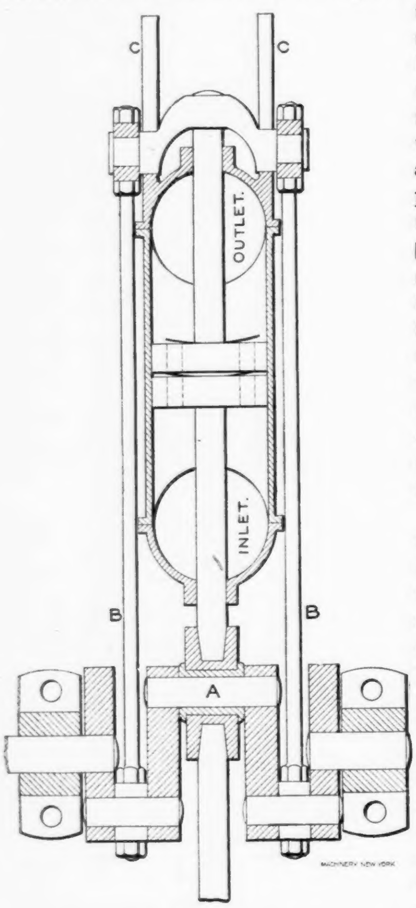
Editor MACHINERY:

After seeing a shop-mate wrestling with a taper tap for a long square-threaded nut, I finally suggested a "step" tap, such as I have used on various occasions, as being more satisfactory than a taper one. He didn't seem to know anything about it and as he liked the scheme after trying it, I thought others might also.

A square-thread taper tap is a sort of a wedge which is cutting and jamming all the way until it reaches the full size. This is likely to bind and crowd to one side.



The step tap is made by turning three or more steps, each cutting a third or quarter of the thread, as the case may be. These steps are usually the length of the nut, so that only a third is being cut at once. The cut shows the tap blank before the grooves are cut. With the exception of the front point there is no taper, a clean, straight cut being made by each size. The writer has had a very favorable experience with this kind of a tap and for square threads prefers it to a taper tap.



While rummaging around some old drawings the other day, I found a description of a double-acting pump, which was simple, easily made and quite a novelty to me. The accompanying sketch makes it clear.

A three-throw crank, the center one working in a Scotch yoke, and the other ones the connecting-rods B B, is the driving mechanism. The pump barrel is a plain cast iron pipe, flanged. The heads are half a sphere, as shown, giving a full opening at each end. This opening is on the side. The top head carries two guides c c for the upper cross-head. The pistons move in opposite directions, being at the center in the sketch. All the valves open as shown.

Supposing pistons to start moving apart, the upper valves close, while the lower ones are open. This draws water through the lower piston at the same time forcing out any water above it.

When the pistons start back, toward each other this action is reversed. The lower piston forces the water about it through the upper piston, which is moving downward, at the same time lifting water from below. This gives a continuous flow of water from the pump.

FRANK C. HUDSON.

* * *

THAT COUNTRY JOB SHOP.

Editor MACHINERY:

I have met men who have taken especial pains to inform me that the "kid" who happens to be fortunate enough (?) to serve an apprenticeship in a job (i. e., principally repair work) shop, stands the best chance to become a first-class machinist (!).

One writer in a late issue of MACHINERY emphasizes this crude "mythologic" nonsense by specifying that the best place for a boy to learn the trade is in a country job shop.

Mr. Editors, that fellow, with those who have preceded him, in such statements are considerably "off" on that line. A country job shop must be one out in the country, and the jobs they get to do must be principally from the country. Country people are largely farmers (one of the most commendable callings) and their machine work must chiefly concern plows, mowers, reapers, and other agricultural implements. The machinery and tools of that class of shops are generally few, poor and worn out; in fact it may be doubted if they were ever much better than apologies for machine shop equipments. The proprietors and the bosses, too, are far from "up-to-date" in practice at the business and for that matter their boasts of theoretical attainments at the business are not very broad.

Now, we are told that these country job shops are just the places in which to become first-class-all-round in early experiences. If our friends really believed this, their home-spun-theory, they would hitch up and drive out to these fortunate shops when they wanted a good man (!) But note, when they come to need good men they think that a good early experience at some machine shop where there was good machinery, good tools, good methods and good work is of itself a good recommendation over and above the crude experiences in crude machinery, tools and crude work. Of course very much depends on the mental caliber of the men who worked as boys in these shops, but mark that boy who has a good make-up for the business. If, perchance, his first experiences happen to occur in one of these country job shops, when he has had sufficient practice to acquire a proper amount of confidence in himself, he gets out, and seeks that shop that has good equipments and does good work. Ah! that fellow has his eye fixed in the direction where he can eventually get good pay for his ability and work.

All honor to that country job shop trying to keep an existence among men by honest labor, but don't tell us "old hands" that they, as a rule, turn out the best workmen; for they don't do any such thing. We had our experiences where good work and good tools were our fortune.

FRANCIS W. CLOUGH.

Springfield, Mass.

* * *

SOMETHING ABOUT OIL-HOLES.

Editor MACHINERY:

Would it not be a good idea for machine tool builders to send out along with their machines a small chart or blue print showing the number of oil-holes and where placed on each machine? In making repairs, I have often come across parts that had probably never been oiled since they were put in for upon taking them apart I have found them "chewed" away to such an extent that I have wondered they worked at all, and upon looking for oil-holes I must confess I have sometimes had difficulty in finding their whereabouts. No doubt when the machines were new and clean the oil-holes could be seen providing one looked close enough and in the right place. They are sometimes in queer places, but when they have been in use long enough to become coated with a thin layer of oil and dirt, or a thick layer as the case may be, the oil-holes which have usually a flush screw are entirely out of sight.

Of course if they were all mechanics who run these machines they would know that there must be an oil-hole somewhere but they are not and if they don't see an oil-hole they have less to oil. I have taken a scraper and exposed to view oil-holes that the man who worked the machine never suspected to be there and in other places where they have never thought or bothered to look for them. Again, a chart could point out the parts that needed oiling often and those that do not, for on some machines there are parts that need oiling every day, and other parts that do not need it more than twice a week, or even once. These charts could be hung near the machines where the men could look them over and a new man's attention be drawn to them. This, I think, would do something toward remedying the haphazard way in which many machines are oiled, or rather left unoiled.

If I have not already taken up too much of your space I would like to go a little farther in the matter of oil-holes, and ask

some of your readers who are practical machinists what is their opinion on having, or not having oil-holes in the lathe carriage to oil the slides? I notice many up-to-date lathes do not have holes for this purpose and to oil them one must put oil on the slides and push the carriage over it, or at least, this is what you are supposed to do. I do not think this is a good way for it stands to reason that the oil will carry more or less dirt in with it, whereas if the oil were let in through a hole on top it would go where needed and working out would carry the dirt out with it, besides there is less waste this way as most of the oil put on the slide is pushed off and must be wiped away as it will make a mess. There is also the annoyance of having to upset one's lathe many a time when quite inconvenient.

Poughkeepsie, N. Y. GEO. H. NORRIS.

* * *

MAKING A MACHINIST.

Editor MACHINERY:

I wish to add a little to Mr. Grant's advice to the youngsters as given in the February MACHINERY. Being a machinist of fourteen years' experience in various shops through the country I have had a pretty good chance to take observations and I would advise a young man, if possible, to work from shop to shop, and to work at different branches of the business. In eight or ten years after he has started at the trade, he will be getting to be an all-around machinist, if he has any mechanical ability about him and has kept his eyes open.

Such men can get a situation almost any time they want one and they make better foremen, than those that stay in one shop all their lives. They will know more about handling men and can see and understand the disadvantages that men are often compelled to work under. I wish there were more all-around machinists. Wages would then be better, because a good man will not stand the abuse that a specialist will. A foreman is not always to blame, however, for the knocks he gives his men, as this lack of adaptability because of inexperience is often a serious matter in the shop, and often keeps a foreman in hot water.

Salem, Ohio. E. E. COOK.

* * *

DRILL RACK—VISE JAWS.

Editor MACHINERY:

Some years ago I desired a way for overcoming the nuisance arising from having often to use small drills the sizes of which had to be determined by gauge.

The idea is not new, though the details have not come to my attention elsewhere.

I had a cast-iron plate made about fifteen inches long, five inches wide and a half inch thick, which I had polished and nickel plated. Then I laid out a series of holes for the different drills, and far enough apart to allow of taking out or replacing drills easily.

Those drills that were most often used had several holes of that particular size so that there were usually enough drills on hand to select the one long enough for a deep hole, or short enough for a center drill, or the one ground for brass or for steel without trouble. Each was drilled to the size of the drill that was to go into it and was numbered for that size. Thus a gauge was made that prevented using a drill larger than the intended one.

At each end of the plate was a pin upon which was hung a drill gauge—one gauge for jobbers' sizes, and one for the number sizes, as the plate held all drills from one-half inch down to number 60.

The nickeled surface prevented rusting and gave a surface that showed the stamped figures well.

There were also screw holes at the corners through which the plate was screwed to the wall with the lower edge standing a little away from the wall to prevent the drills working out by the jar of the building.

This proved much more satisfactory than a modern block, as the holes did not wear enough to allow a drill to be placed in a smaller hole than the proper one, and any amount of shake at once indicated that a drill had been replaced in a larger hole than the proper one.

Another plan I always liked very much was one I used for vise jaws.

I made a pattern that fitted over the jaws of the vise loosely enough to allow for shrinkage and roughness of castings, and had sets of copper and lead jaws cast. They were much neater than those made by bending sheet metal into shape and were much more easily renewed. When the copper jaws became hard by use, they were annealed by heating and quenching. FREDERICK M. BUSH.

* * *

EPICYCLIC BACK GEARS.

Editor MACHINERY:

In your February issue we notice an article on our "Patent Internal Gearing as applied to Drill Heads, and we are desirous of correcting your impression that the gear would be more noisy than ordinary spur gearing . Our experience is that it is much quieter; probably owing to the fact that the wheels all run in the same direction; the teeth are longer in gear and more at one time are in gear, owing to the flat bevels; and the constant lubrication from the wheels dipping in the oil bath at each revolution. As a proof of this we may mention that we have working in Bristol a portable electric driving gear consisting of an alternating current motor, a set of reduction gears and a dummy pulley arranged on a base plate. This motor gives 4 horse power at 2700 revolutions per minute, and the set of gearing reduces the speed at the pulley to 210 revolutions. At this extremely high speed the gearing makes no more noise than the motor itself, and works very efficiently.

HUMPAGE, JAQUES & PEDERSEN.

[In the description referred to we expressed the opinion that the back gears made by Messrs. Humpage, Jacques & Pedersen, which consists of a set of epicyclic bevel gears encased in the belt cone of the machine, would be more noisy than some of the simpler American types of spur gearing operating on the same principle. If actual test shows the reverse to be true, there is nothing more to be said except to compliment the designers upon their excellent workmanship. We understand that this gear has been patented in this country and we trust that there will soon be an opportunity to see it in operation here, thus affording a better chance to justly estimate its running qualities. —ED.]

* * *

NOTES FROM FIFTY YEARS OF ENGINEERING EXPERIENCE—COMPLIMENTS FOR "MACHINERY'S" CONTRIBUTORS.

Editor MACHINERY:

The March number of MACHINERY contains several very interesting articles, which come so fully into the field of my own experience, and the conclusions of which agree so fully with my own opinions, as derived from such experience that I desire to endorse them, as appearing to me to be entirely correct.

The first one to mention is on "Prony Brake Test Errors," and is signed "F. E. R.," in which the writer refers to a previous article by "Bell Crank," and shows the fallacy of the argument that the weight of the brake itself produces any material effect on the friction recorded by it, and that such as it does can be easily eliminated by properly balancing the weight of the brake lever.

In a previous article in Machinery on "Dynamometer," I have shown a simple straight-lever brake, like the illustration No. 2, in the article referred to, in which the weight of the "scale pan," at one end was exactly balanced by the weight of the piston, in the "dashpot," at the other, and in such cases, where the brake was so heavy as to produce any essential increase of load on the journals, where as "F. E. R." says, its only effect is felt, I have suspended the brake beam at the center of gravity, from above, either by the chains of a "differential block," or by a rope, with a strong spiral spring connection, so that the spring would just allow the brake to touch the pulley, and this can usually be very easily done. At the turbine tests at the Centennial in 1876, where the shaft was vertical and where the weight of the brake would have produced objectionable step friction, I suspended the whole weight of the brake by differential blocks, as noted, so that it would not bear on the lower rim of the flanged pulley, and I do this in all cases of tests of vertical shafts, allowing that the weight of the friction pulley is fairly offset by the weight of the usual bevel gear, which it replaces. I always make a point of balance. My brake levers are placed over the center of the shaft in the first place, and then if they are as heavy as suggested by F. E. R., I suspend them from that center in one of the ways above noted. I never bear down on a platform scale, but lift a dead weight if possible.

the "chatter" incident to friction, makes the scale beam too "joggy."

The article immediately following the one already referred to is by Prof. C. H. Benjamin on "Starting and stopping a boiler test," and here also, I wish to express my entire agreement with, and approbation of his conclusions. Among my very earliest engineering experiences more than half a century ago, or early in the "40's," before many of our present experts were born, was that of assisting Dr. Samuel L. Dana, the chemist of the Merrimac Print Works in Lowell, Mass., in a series of experiments on the combustion of fuel, the evaporation of water, the design of boilers, and the mode of boiler setting, and during the 50 odd years that have followed, I have conducted a great many similar experiments, resulting in the conclusion, that all this elaborate nonsense of raking out the fires, weighing the kindling wood, and starting fresh fires, was a complete farce, and that the only sensible and accurate method was that recommended by Prof. Benjamin, of starting the observations with everything in full operation, noting the height of water in the boiler, and fire in the furnace, and taking care to have both in exactly the same condition at the end of a 10 or 12 hour test. This gives results which will exactly compare with those attained on an ocean steamer in an Atlantic passage, or in a manufacturing establishment, where the boilers are often run continuously from Monday morning until Saturday night. Something over thirty years ago I carried out a series of tests of the evaporation value of different coals, Lehigh, Pittston, Broadbrook, Lackawanna, etc., etc., for two consecutive winters, in which the fires were only "banked" at night, during the week, and the whole consumption, as well as the measure of the water supplied to the boilers, was noted for the entire six days, and the same experiments also covered a series of boiler tests embracing plain cylinder, return flue (two flues), horizontal tubular and locomotive types, with the result that the Lehigh coal and the horizontal tubular boilers came out ahead. Prof. Benjamin has entirely covered the ground, and I have only to repeat, that I endorse him fully.

The next article to strike my attention is also by Prof. Benjamin, and here again, he touches on ground which I have traversed and retraversed for many years. This article explains the conditions of the tests of the power consumed by shafting, which he reported in a paper read before the American Society of Mechanical Engineers some two years since, and showing that these tests were mostly made in machine shops where heavy tools, scattered at long distances apart, required an excess of shafting to drive them, and suggested the economical application of electric transmission.

In this they differed from my own experiments, which have been generally made in large cotton and woolen mills where a great number of machines, requiring in most cases but a small amount of power each, were placed as closely together as possible, to admit of the easy transmission of the material from one process to another, and the amount of shafting required was consequently reduced to a minimum. His lightest results, of about 15 per cent. agreed very closely with my own notes, which I again sub-divide, into 10 per cent. for the actual shafting, and 5 per cent. for the machine belts, which latter item, I have usually charged to the machine itself. Five per cent. for this item is perhaps, rather a small allowance, as in many light machines it would be much higher. There is however, one test mentioned in this article, to which I was particularly attracted, and that was the test of a line shaft, 90 ft. long, 1 15-16 in. diameter, and making 200 revolutions per minute. This agrees so closely with my experience, that I long since deduced a "rough-and-ready" formula for estimating the power required for shafting, which is 1 h.p. for each 100 ft. in length, of 2 in. shaft, at 200 revolutions per minute, if the shafting were properly in line and lubricated. Nearly all of his weighings of machine tools have been of a heavier class than those to which my attention has been called, and which have been in most cases the small tools used in cotton mill repair shops, or in the manufacture of small machines, lathes, drills, etc., etc., which I usually average in a rough estimate, on small work at about 3 tools to a horse power, or on very light work at 4 per horse power, not including the countershafts. In endorsing this article also, it seems to be about enough on the complimentary line to-day, unless it is to express my appreciation of Mr. Van Dervoort's, "Shop Talks for Young Mechanics."

SAMUEL WEBBER.

"CHERRY RED."

Editor MACHINERY:

A saying that used to go the rounds years ago among the smiths and tool makers and other supposed-to-be wise sages in the steel art was, that in heating a piece of tool steel (then called cast steel) for forging or tempering it should not be heated beyond a "cherry red." This standard for the color of steel when hot was supposed to be a safe limit to prevent overheating or burning, but was it, or is it now? Suppose we investigate this "cherry standard" a little. There is almost an endless variety of cherries and corresponding difference in their colors, from the white fruit with a tinged spot on one side to the black variety. The white red, the pink red, the bright red, the deep red one, says our cherry friend. Yes, but that is somewhat unred one says our cherry friend. Yes, but that is somewhat uncertain for degrees of heat, and I confess that this expression "cherry red" is about as definite as that of a red apple.

In later years some of our American tool-steel makers have used another standard of color to protect their high grade steels from being overheated, namely, a "blood red." Blood of healthy creatures is quite uniform in color and thereby becomes a very much better standard of color for heating high grade steels. Of course degrees of heat is the true theoretic standard; but the common user of steel has not the means (at present) to thus "take a heat," so he has to trust to his eye, guided by a uniform standard of color, and perhaps there can be no better one than that of "blood red."

It commended itself to me at once as very much better than "cherry red." Some low grade steels do not suffer from a greater heat than blood red, but they haven't the business in them that a good high grade steels have, and high grade steels must be treated right or they will not answer the purpose.

Springfield, Mass. F. W. CLOUGH.

* * *

The difference between the action of an Otto gas engine and a Diesel motor is thus described in "The American Engineer:" The first stroke of the Otto gas engine draws a mixture of air and gas into the cylinder, the second stroke compresses it, at the beginning of the third stroke it is exploded, and during the third stroke the piston is driven forward as a result of the explosion, while the fourth stroke expels the burned gases from the cylinder. The operation of the Diesel motor is entirely different. The first stroke draws air into the cylinder, the second compresses it to about 600 pounds per square inch, at the commencement of the third stroke oil is forced into the cylinder against the high pressure by means of a small auxiliary pump driven by the engine, and is immediately exploded by the heat generated by the compression, the piston is driven through the third stroke by the force caused by the burning oil, and the product of combustion is expelled during the fourth stroke. The air compressed by the pump referred to is also used in starting the engine, and this gives the Diesel motor an advantage over others by being always ready to start. The advantages of this motor have been summed up as follows: Simplicity, specially as regards regulation, automatic ignition of the gas without requiring any mechanism for this purpose, regularity of operation under varying loads, good design of oil injecting device, whereby the oil is completely consumed, extreme cleanliness, and the ability to start instantly at any time. The consumption of petroleum, according to Prof. Schroter, given in a paper last year before the Institution of Civil Engineers (England), is 0.524 pound per brake horse power and 0.396 pound per indicated horse power per hour.

* * *

According to the "Scientific American," the horseless carriage is not such a recent fad as we have been led to believe. It states that the mechanic Vaucanson was honored, in 1740, by a visit from Louis XV. for the purpose of inspecting the carriage which ran without the aid of a horse or other visible means of propulsion. Two persons in the vehicle made the round of the courtyard to the satisfaction of his majesty and suit, but, though a promise was secured of royal patronage, the Academy of Sciences declared that such a conveyance could not be tolerated in the streets, so the scheme had to be abandoned. The motor power was supplied by a huge clockspring, so that only a short journey was possible, but the gear seems to closely resemble that of the horseless carriages of to-day.

PRACTICAL PROBLEMS.—6.

PROBLEMS 11 AND 12, WITH ANSWERS TO PROBLEMS 9 AND 10.

FRED E. ROGERS.

11.—To find the Position of Equilibrium.

The pulley S has two holes drilled through the rim midway between the junctures with the arms and through the holes is placed the steel bar E having on its end the cast iron ball B. Attached to the rim D at a distance from the center of 16" is the cord supporting the weight C. It will be observed that the point of attachment of the cord is directly on the center line of the pulley arm G, and that the vertical line of the cord passes over the corresponding point in the rim at F for the pulley arm H.

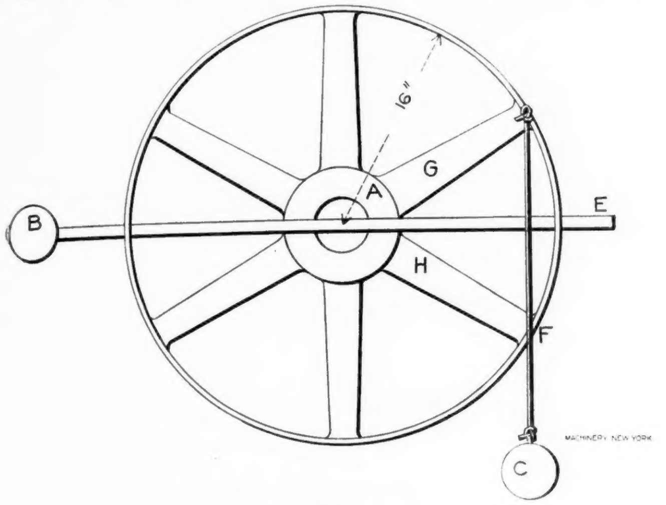


FIG. I.

The length of the bar E is 69½", with a diameter of 2", the diameter of the ball B being 5". The diameter of the ball C is 6", and neglecting the weight of the cord, we wish to find the position which E must occupy with the other members in the position shown, so that the whole will be in balance. It is understood that the pulley is balanced when free from the bar and weights. Take the weight of cast iron to be 480 pounds per cubic foot, and steel as 490 pounds.

12.—Wanted the Position of O and the Pull on Cord K.

The rectangular cast iron block I has a length of 9", a breadth of 5" and a height of 4". It lies as shown on the plank L, and has projecting from its upper surface the rod J, which is located 3" back of the face M. Taking the co-efficient of friction to be .20, and neglecting the weight of the rod J, at what point must

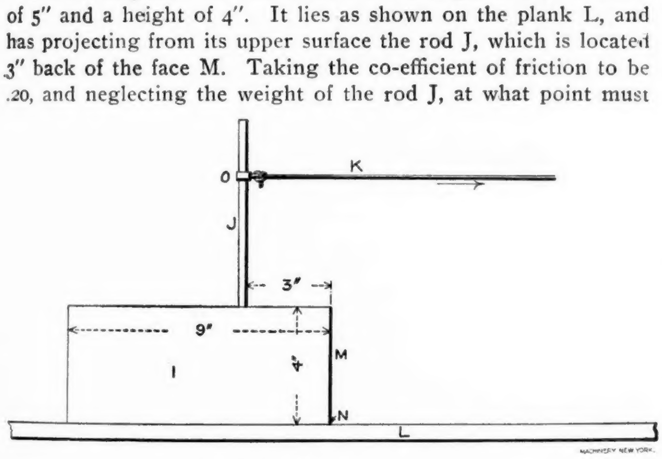


FIG. 2.

the ring O be placed on J, so that with a pull along the cord K the force tending to slide the block and that tending to tip it on the edge N will be equal? Also, what will be the pull on K when these forces are equal?

ANSWERS TO PROBLEMS 9 AND 10.

Mr. C. Albert Wettengel, of Pittsburg, Kan., sends solutions to problems 9 and 10, which follow:

Problem 9.—Considering the belt as a positive means of transmitting motion, as, for instance, some style of link chain, and the diameters as pitch diameters, then the following equation,

$\frac{100}{62832}$ = part of revolution 7 P, 14 T wheel makes,

$$\frac{\frac{100}{62832} \times 64 \times 24}{\frac{36}{24}} = \frac{\frac{6400}{2618}}{\frac{36}{24}} = \frac{6400}{3927}$$

gives the ratio of the pulley diameters, the intermediate result showing that Y makes more turns that X, hence representing diameter of X by 6,400, the diameter of Y is 3,927.

With shaft A turning, as shown, one of the worms must be right hand and the other left hand, to move the rack down.

Problem 10.—To solve this problem, suppose that wheel M is held from turning, while the internal gear D and with it the frame of the hoist is given one rotation, then

$$\frac{56 \times 32}{12 \times 12} = \frac{112}{9} = 12\tfrac{4}{9} \text{ the}$$

number of turns wheel S makes, while D turns once, but when D is stationary, S must also turn once to make up for the rotation of D, in the first place, because the number of rotations of S is relative to D. Hence S makes 12 4-9 + 1 = 13 4-9 rotations to one of M.

We have received the following solutions from Mr. Elmer G. Eberhardt, of Newark, N. J., which are in his usual clear style:

Problem 9.—As the rack is to move 1-100th of an inch for one turn of spindle A, we must first find the part of rotation the rack pinion must make to feed 1-100th of an inch, and as the pinion is 14T. 7P., its pitch circumference is 6.2832", and one turn of this pinion moves the rack 6.2832"; hence, we find that to feed 1-100th of an inch, the pinion must turn 1-62832 = .00159154 turns for one turn of spindle A. Now, letting x-y equal required ratio, and equating all the factors of the feed mechanism, we have

$$\frac{36}{24} \times \frac{x}{y} \times \frac{1}{24} \times \frac{1}{64} = .0015915 \text{ whence}$$

$$\frac{x}{y} \times \frac{1}{1024} = .0015915$$

$$\frac{x}{y} = 1.6296 : 1$$

or, in round numbers, the required ratio is Ans. x:y::13:8, nearly.

Problem 10.—Let us first move the internal gear one turn in a minus direction, which movement (supposing the frame E to remain stationary) will give A $\frac{56}{12} \times \frac{32}{12}$ turns in an opposite direction, which is in a plus direction, but now the internal gear is to be turned back one rotation to its orginal position, carrying with it the frame E and also A, thus giving A one more turn, making $(\frac{56}{12} \times \frac{32}{12}) + 1$ turns of A for one rotation of M, which is 13 4-9 turns of A = 1 of M.

Mr. D. W. Randall, of Manhattan, Kan., says that he finds the problem department very interesting, and submits his solution to No. 9:

Problem 9.—7 diametral pitch equals .448 circular pitch; 14 x .448 equals 6.272" equals circumference of pitch circle of the pinion engaged with the rack. If shaft A makes one turn for each .01" drop of rack, it will make 6.272 ÷ .01, or 627.2 turns for one rotation of the rack pinion. In one turn of rack pinion F makes 64 revolutions, and Y makes 24 x 64, or 1,536 turns. While A makes 627.2 turns, X makes 36-24 x 627.2, or 940.8 turns. Then the number of turns of X is to the number of turns of Y as 940.8 : 1,536, or as 49 : 80, nearly. And the diameter of X : the diameter of Y = 80 : 49, or practically 8 : 5. Worm E should be right hand and worm F left hand, or vice versa.

Mr. Charles Stecher, of Chicago, Ill., says in his solution of problem 9 that the worm F should be left hand and worm E right hand.

Problem 9.—One turn of the 14T. pinion will advance D 6.2832" and will require 64 rotations of F, since there are 64 T in the worm wheel, as many rotations of the worm being required as there are teeth in the worm wheel; 1,536 rotations of Y will be required to advance D 6.2832", which is found by multiplying 64 T x 24 T, since it takes 24 turns of E to turn its worm wheel once, and to run with the shaft of F it will require to turn 64 times as fast.

1536 rotations ÷ 6.2832 = 244.46 rotations of Y to advance D 1 inch.

244.46 × .01 = 2.44 rotations of Y to advance D 1-100".

36 T ÷ 24 T = 1½ rotations of X to one of A.

1½ : 2.44 = y : x.

x = 1.63

y = 1.

The following answer to problem 10 is given by Mr. W. Boley, Hartford, Conn. Mr. Boley says:

Problem 10.—Suppose E to be stationary, but D movable, then one rotation of D will give pinion A

$$\frac{56 \times 32}{12 \times 12} = 12\frac{4}{9}$$

but since D is stationary, the frame E will turn in the same direction as the pinion A, which will give B and the frame E a reverse rotation, so that the effect produced by A necessitates one more rotation of A, with the frame E, thus giving 13 4-9 rotations of S to one of M.

Having become interested in "Practical Problems," Mr. Fred S. English sends a solution to the cone-pulley question.

Referring to the sketch one rotation of A will rotate X 36-24, or 1½ turns, and since the rack pinion is 7 P. with 14 teeth, the pitch diameter is 2". Hence, it must move $\frac{.01}{\pi 2} = \frac{1}{628.32}$ of a rotation, while the rack moves 1-100th of an inch, and since F, and therefore E, rotate 64 times to one turn of the rack pinion, they will make 64-628.32 rotations. Y turns 24 times to one turn of E, hence will make $\frac{64 \times 24}{628.32} = 2.4446$ rotations.

Then the ratio of X to Y will be $\frac{x}{y} = \frac{2.4446}{1.5} = 1.6297$, or X must be 1.6297 times as large as Y. One worm must be right hand, and the other left hand.

Mr. Thomas F. Hannan, of Schenectady, N. Y., also sends a correct answer to problem 9.

The solution here given for No. 9 is from Mr. E. L. Darling, Providence, R. I:

Problem 9.—If A makes one rotation, then the shaft which is geared to it will make $36 \div 24 = 1\frac{1}{2}$ rotations. Since the rack moves down 1-100th of an inch to one turn of A, the gear will turn through an angle corresponding to an arc on the pitch circle, whose length is 1-100th of an inch. In order to find out what fraction of a complete rotation this corresponds to, it is necessary to find the pitch diameter of the 14 T. gear, which will be $14 \div 7 = 2''$ From this the circumference of the pitch circle is found to be $3.1416 \times 2 = 6.2832''$. Hence, the fraction of a revolution made by the gear will be

$$\frac{1}{100} \div 6.2832 = \frac{1}{628.32}$$

The worm F makes 64 revolutions, while the gear makes one Hence F makes $\frac{1}{628.32} \times \frac{64}{1}$ and since miter gears transmit the number of rotations unchanged from one shaft to the other, the worm wheel at E will also make 64-628.32 of a revolution.

The worm wheel at E will make $\frac{64}{628.32} \times \frac{24}{1} = 2.44$ revolutions.

Then diameter of X : diameter of Y :: 2.44 : 1½. That is, X should be 1.63 times the diameter of Y. The worm F should be right hand, and E should be left hand, or vice versa.

ABOUT PROBLEM 7.

Mr. James Dangerfield, Elgin, Ill., says:

We were neatly caught by problem 7. You say truly, "it is doubtful if any practical pitch can be found that will satisfy the requirements." For fear it may prove so, I submit a device that

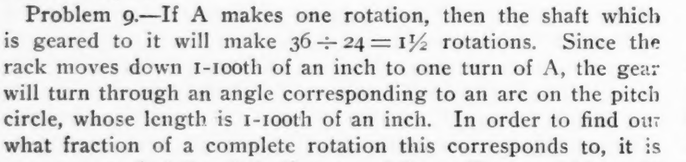


FIG. 3.

may be one way out of the difficulty, which is shown in sketch, Fig. 3. Make rim A a little over twice the depth of wheels D and B. Also make the intermediate wheel double, the two parts C and G being cut of the same pitch, and number of teeth as D and B; G to be clamped to C by the nut H. Arrange D in mesh with C, and B in mesh with G, and put the wheels together with the nut H loose, so they are free to adjust themselves to the proper position. Now tighten the nut H, and there you are.

The problem published in the February MACHINERY, relative to the ratios of the internal and pinion gears, presents some unusual difficulties, and as no direct solution has been received, the following approximate solution may be of interest, and will be found sufficiently accurate for all ordinary requirements.

The angle G L F of Fig 4 having been found by reference to a table of sines and tangents to be 53° 7' 48", we can readily find the other angles of the triangle L G F, since two sides are radii of the same circle, and we already know the angle opposite one radii. Thus angle F G L is also 53° 7' 48" and angle K F G is 106° 15' 36", since the exterior angle of a triangle equals the sum of the two opposite interior angles.

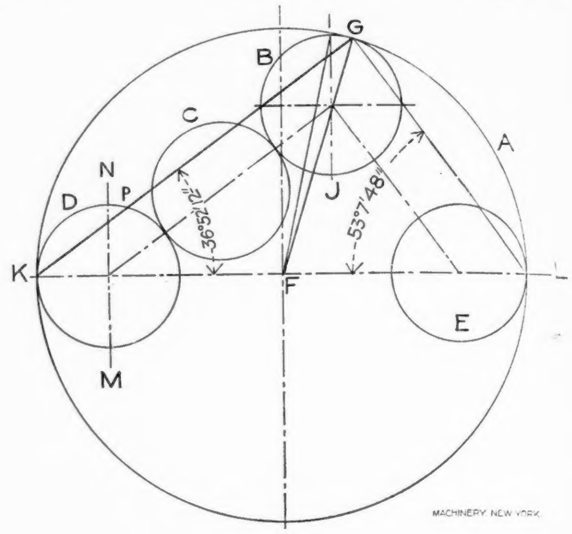


FIG. 4.

Supposing the center of a tooth space of the internal gear A to be at G, the center of a tooth of pinion B must be on the radial line G F and the line K L must also cut a tooth and tooth space in pinion D and gear A, into equal proportions, at K. Under these conditions we must select a number of teeth for A, which is divisible by 7, since it was found that the pinions are two-sevenths of the diameter of A, and the number must also be one that, together with its two-sevenths part, will each, when multiplied by $\frac{106° \ 15' 36''}{360° \ 00' \ 00''}$ give either whole numbers for results or numbers having fractional parts, which are equal.

The reason for this is that points in D and B always keep the same relative position, and thus point P in pinion D corresponds to G in pinion B. Therefore arc K P subtends the same angle as arc K G or 106° 15' 36".

106° 15' 36" = 382,536"
360° 00' 00" = 1,296,000"

$$\frac{382536}{1,296,000} = \frac{1771}{6,000}$$

Selecting 147 teeth for A we have for the pinions $\frac{147}{1} \times \frac{2}{7} =$ 42 teeth.

$\frac{147}{1} \times \frac{1771}{6000} = 43.3895$ tooth spaces from G to K.

$\frac{42}{1} \times \frac{1771}{6000} = 12.3970$ teeth from P to K.

The fractional parts are nearly equal, showing .0075 difference of tooth and tooth space, which is equivalent to a difference of only .0014" in the proper positions of the opposing teeth in A and D at K.

Unfortunately, Mr. S. Moe, of Chicago, Ill., delayed his solution to No. 8 until too late for the March issue. We regret not having space to present it this month.

* * *

A Western newspaper contains the information that the torpedo boat built by Moran Bros., of Seattle, exceeded her contract speed 1.62 knots, making 28.97 knots per hour, or 33.36 miles. This, and the unusual record of the Oregon, built by the Union Iron Works, of San Francisco, calls forth the remarkable statement that the superiority of the iron ships built on the Pacific Coast, is possibly due to the climatic conditions. The different parts are subjected to nearly the same temperature the year round, and the result is that better fits and alignment result. The superiority of these vessels is, however, we think, due to good designing, and first-class workmanship, rather than to the influences of the weather.

MAKING A PIECE STRONGER BY MAKING IT WEAKER.

This expression, if we remember his words correctly, is due to Prof. John E. Sweet. Some time ago, at one of the meetings of the American Society of Mechanical Engineers, he was discussing the strength of piston-rods, and suggested that they might be made stronger and less subject to fracture by making them weaker. The point that he wished to emphasize is brought out in Fig. 1, which shows a piston-rod with a section of the cross-head. The rod is drawn up against the shoulder h, and experience shows that this shoulder is a source of frequent breakage. Every time there is any deflection of the rod, due to the rocking of cross-head, when it passes the center, or to the engine guides and cylinder being out of line with each other, the effect is concentrated at the shoulder, and these repeated stresses may in time cause the rod to give way. Prof. Sweet's suggestion was to turn the rod smaller, as indicated by the dotted lines, thus bringing the bending action at some point in the body of the rod, as at b. If this were done, the rod would have been made stronger, in effect, at least, by making it weaker, assuming, of course, that the diameter of the body were large enough to ensure stiffness.

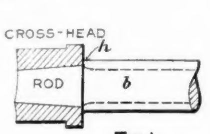


FIG.1

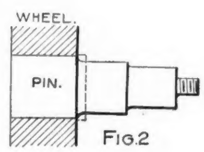


FIG.2

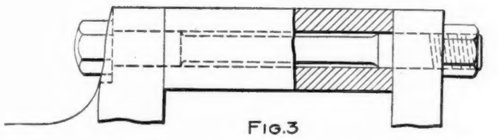


FIG.3

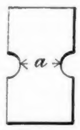


FIG.4

FIG.5

FIG.6

FIG.7

FIG.8

There are other instances where this same principle may be applied with good results, and there are plenty of places where it ought to be applied, but where it is not. In Fig. 2 is a representation of a locomotive crank-pin, and it is very common in modern practice to find the part of the pin that is forced into the wheel larger than the body of the pin. Formerly the practice was to make the body of the pin larger than the other part, and to force the pin into the wheel, until it came against the shoulder. Again, experience shows that it is safer to keep the result of the bending action away from the point where the pin enters the wheel, and if a shoulder be thought necessary, it would seem to be a common-sense plan to make it as shown by the dotted lines, still having the body of the pin smaller than the part that forces into the rod.

The application of our principle, however, is not alone found in machine parts where bending occurs. In Fig. 3 is shown a bolt which is in direct tension, and yet which is actually made more serviceable by turning it below size at the center. This bolt is supposed to hold together the jaws of the crank end of a connecting-rod, and to hold the distance piece in position, thus making a solid and sure support for the crank-pin boxes. When it is desired to remove the boxes, the bolt is withdrawn, the distance-piece taken out and the boxes can be slid out at one side. With every vibration and movement of the rod there are repeated stresses occurring in the bolt, which tend to strain it at its weakest point, which is in the screw thread. These stresses are in the nature of repeated shocks of varying intensity, and as the grooves of the thread are narrow and sharp at the bottom, there is a very short length of metal to sustain the strain and a break at this point is not uncommon. With the bolt turned smaller at the center than the diameter at the bottom of the screw thread, this will be the weak part of the bolt, and being of considerable length, it will have ample elasticity to withstand the shocks brought upon it, and thus relieve the screw thread of the strain.

While the sharp grooves of the screw threads are not well adapted to withstand stresses that are of the nature of a shock, it sometimes happens that properly shaped grooves will, under a steady tensile stress, show the opposite effect, and here we have a case where pieces may be made stronger per square inch of section by making them smaller.

For example, some tests upon boiler joints made for the United States Government at the Watertown Arsenal, showed that a plate, which ordinarily was capable of withstanding a tensile stress of 53,000 pounds, actually broke at a stress of over 73,000 pounds per square inch between the rivet holes, the rivets themselves being so large that the plate broke by tearing between them, instead of by shearing the rivets. Fig. 4 shows a piece of the plate cut from between the rivets, and Fig. 5 a strip of plate of a width, a, equal to the distance between the edges of the rivet holes in Fig. 4. The plate in Fig. 4 would be expected to show under test a considerably greater strength per square inch than the plate in Fig. 5, owing, no doubt, to the fact that the successive layers of metal tend to reinforce one another. While the grooves do not make the piece itself stronger, they do make it stronger per square inch.

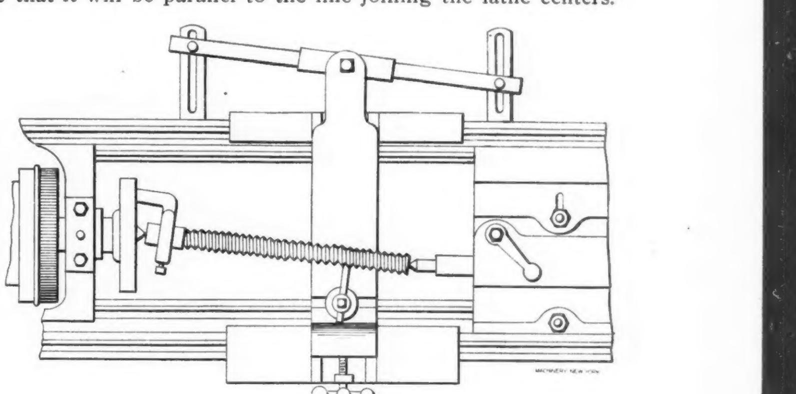
For this reason, round test pieces should be made, as in Fig. 7, rather than as in Fig. 6, for if grooved, as in the latter figure, as they often are, the results of the tests are sure to be misleading.

A word should be said, however, about grooves, like those of screw threads, as shown in Fig. 8. If the groove resemble a V thread it will probably prove to be a weak point, and the tensile strength at the bottom of the thread may be less per square inch, even if there is no shock or jar, than would be shown by a test on the body of the piece. But if the groove correspond to the United States standard thread, as indicated in the upper part of the figure, the strength per square inch will probably not be diminished, if we may take as evidence the following extract from Prof. Forest R. Jones' new book upon machine design. He states that "It has been found by experimental investigation that, on account of this strengthening effect of the thread, with fairly good rubbing surfaces, reasonably well lubricated, the axial tensile stress per square inch which will cause rupture in a United States standard screw bolt, while tightening the nut, is practically the same as the breaking strength per square inch of the body of the bolt."

* * *

TO CUT FRACTIONAL THREADS.

The apparatus described in problem 8, of the February MACHINERY for cutting odd pitches of thread has called from R. E. Flanders of Providence another method of reaching the same result in a simpler way. He says that his plan consists of merely setting over the tail stock of the lathe to any desired angle, as shown in the sketch, and setting the Slate taper attachment so that it will be parallel to the line joining the lathe centers.



"Now the lathe might be geared so that the threading tool would move one inch along the center line of the lathe while the spindle made eight revolutions, but the point would traverse a greater distance by reason of following the incline of the work being threaded, so that a coarser thread would be cut proportionate to the angle between the axis of the threaded part and the axis of the lathe.

In using this method, gear the lathe to the nearest possible pitch, finer than the required fractional pitch. Then the appended formula will give the angle of the taper required.

P = required pitch in threads per inch.
p = nearest finer pitch.

$$\text{Cosine of the angle} = \frac{P}{p}$$

This method introduces an error similar to that encountered in indexing taper pieces on the spiral head of a milling machine, namely, the varying distance between the point of contact of the dog and the face plate slot, and the center line of the spindle. This irregularity, however, does not affect the pitch of the thread, but the regularity of the twist only and, therefore, is of no practical account under ordinary conditions, in fact, the error is practically incommesurable.

On mentioning this plan to a acquaintance, I was told that the idea was not a new one, but, as I have never seen it in print, I offer the scheme for what it is worth."

* * *

ITEMS OF MECHANICAL INTEREST.

A RAILWAY APPLIANCE.

The interesting device which is here illustrated has recently been placed on the market under the name of a spring dampener. The object of it is to produce as smooth effects with coil springs under freight cars and other rolling stock as is obtained with the more expensive elliptic form. With the latter the friction between the adjacent leaves is an important factor in preventing the sharp vibration, which is such a disagreeable and destructive feature of the ordinary coil type. To enable the coil spring to produce the same effect as the elliptic form, this dampener is made of the form shown.

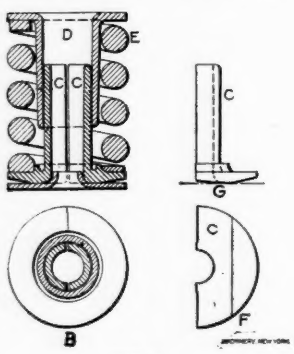


FIG. 1.

It will be seen that the spring E is held between the flanged piece D, and the flanges of the semi-circular pieces C C. The latter are beveled on the bottom, as shown at F and G, so that they form a pair of rocking levers, with their fulcrums on the flat part bearing on the lower plate. The load transmitted through the spring tends to spread these levers apart, but they are held in place by the sleeve D, so that a certain degree of friction is caused by the rubbing surfaces. With proper proportions of parts, results are claimed which equal those obtained with elliptic springs, and at a greatly reduced cost, together with a saving of space, which is often a desirable feature.

A TOOL FOR BICYCLES.

The wheelman having a mount with the old, but reliable, crank fastening, consisting of the taper pin, with a nut, will appreciate the handy device shown herewith, which is being manufactured

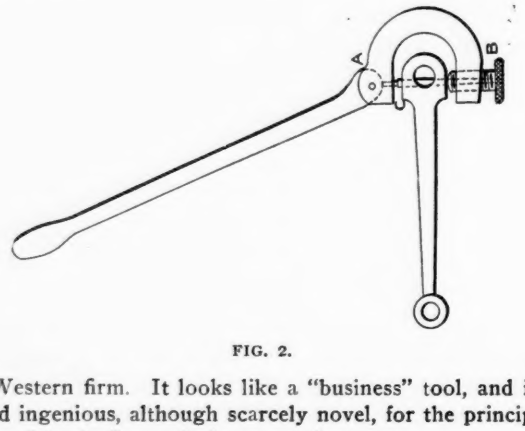


FIG. 2.

by a Western firm. It looks like a "business" tool, and is simple and ingenious, although scarcely novel, for the principle has been used in similar tools for some time past.

BELTING FOR CONE PULLEYS.

This belting is of English manufacture, and as shown in the accompanying Fig. 3, is designed for use on true cone pulleys which are often desirable, but are destructive to ordinary belting, and usually unsatisfactory. The upper view shows the style made for open belts, and the lower view that which is to be used in crossed driving. It is stated in the descriptive circular that the taper of the pulleys should not exceed 10 per cent. of the width of face, and that with crossed driving more satisfactory results are likely to be obtained than with open belts, as the an-

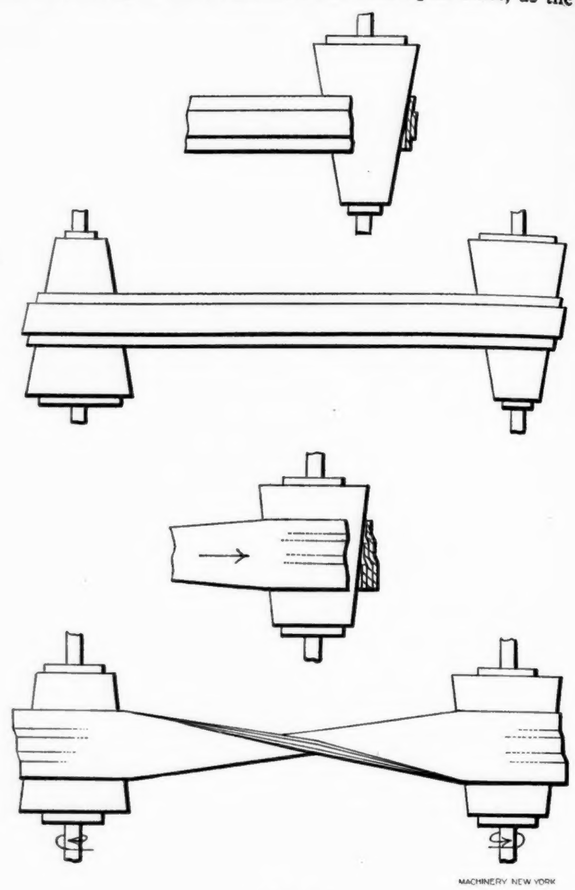


FIG. 3.

gularity of the cone is counteracted by the thickened layers of the belt, which thus produce a uniform circumferential velocity on each edge of the belt, together with equality of pull.

In Fig. 4, about the same principle is employed in quarter-turn driving, and, owing to the diameter being increased by the

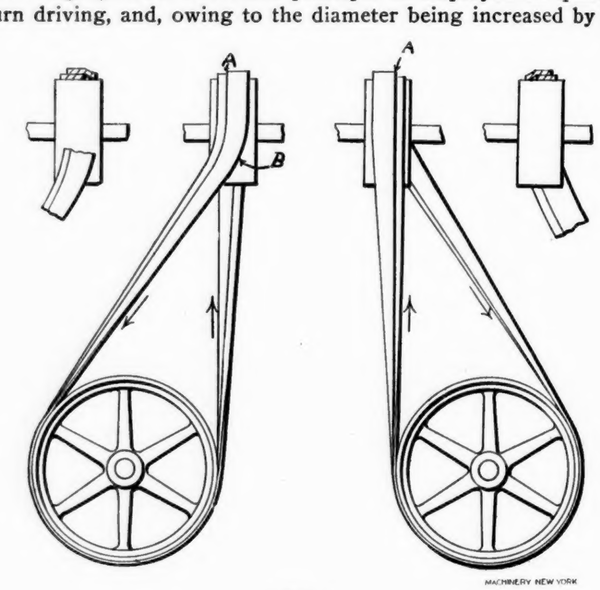


FIG. 4.

under layer, the heaviest strain on the belt comes at A, instead of at the edge, B, as it does with ordinary belts. This, together with the fact that a given width of belt will give greater driving effect than the ordinary form, will undoubtedly prove of advantage in many places.

* * *

The days of the wooden ship are not entirely over, it seems, as Arthur Sewall & Co., of Bath, Me., have recently completed the four-masted wooden ship Susquehanna, having a total depth of hold 19 feet 1 inch. The gross tonnage is 2,745, and net tonnage 2,629

HOW AND WHY.

A DEPARTMENT INTENDED TO CONTAIN CORRECT ANSWERS TO PRACTICAL QUESTIONS OF GENERAL INTEREST.

Give all details and name and address. The latter are for our own convenience and will not be published.

81. W. D. writes: In the place where I work there is a 12 H.P. semi-portable horizontal engine and locomotive boiler. The coal used is steam lump, and it is evidently of very poor quality, for the grate becomes covered with a substance like clinkers which is hard to get off. A great many clinkers are also formed, which have to be taken out with the shovel, as they will not pass through the grate. From the boiler to the chimney is a stack eight or ten feet long. The chimney is of brick, about 35 feet high, with 12 feet of 12-inch stack on top. The boiler has 40 tubes, two inches in diameter, and when they are clean there is an unusually good draft; but they clog up very quickly, and have to be scraped four or more times a day. The stack takes fire once or twice a week when the flues are clean, and there is a blazing fire on the grate. Can you explain the cause of the trouble with the tubes, and suggest a remedy? 2. Will a little salt, mixed with the coal, prevent the grate from choking? A. The diameter of the tubes is too small for the kind of coal that you are using. Three-inch tubes would be better, and the total tube area should be about one-fifth the area of the grate. Bituminous coal gives off a great deal of smoke, which will at best form a deposit in a short time. When the tubes are small the deposit is heavier, because they deaden the flame passing through them, and combustion is not complete. The principle is the same as in holding a piece of sheet metal over a lighted candle. If kept away from the flame there will be no deposit; but if it be brought in contact with the flame, it will check combustion and the plate will become smoked. From the fact that your chimney burns out so frequently, it is probable that there is also a heavy liquid deposit coming from the condensation of some of the volatile gases, which forms an inflammable soot. The clinkers show that the coal is of poor quality. The best remedy is to change the kind of coal. If this be impracticable, we should recommend a reduction in the size of the grate, and the introduction of forced draft, either by means of a steam jet, which draws and forces air into the ash-pit, on the plan of the injector, or by a fan-blower. Forced draft enables a low grade of fuel to be used, and tends to make combustion more complete. It also makes a more rapid passage of the smoke and gases through the tubes, which assists in keeping them clean. We do not care to offer suggestions for the proportions and arrangement of such an apparatus, however, without more complete data. 2. You can easily determine by trying, but we see no reason why it should have this effect.

82. H. A. S. writes: I find it necessary in my business to remove the scale from forgings of wrought iron and steel. Can you give me a formula for a preparation which will leave the forgings bright and clean? Give me, if you please, the quantity of acid to be used, the quantity of water, and the kind of a vessel it will be necessary to have to keep the solution in. A. This question was referred to Mr. David Gorrie, instructor of forging and metallurgy, English High School, Chicago. He says: "Use sulphuric acid, three parts, to one of water, and have a wooden tank, lined with lead. The quantity of the solution can be varied to suit the quantity of work. The latter should be slightly heated when taken out to keep from rusting."

83. J. P. S. writes: I am in charge of an elevator run by hydraulic power. Why do the sheaves revolve at different speeds as the cables pass over them? 2. Why will a steam pump, operating with 80 pounds steam pressure, pump against a water pressure of 120 pounds, or more? Also, if there were no pump governor, how high would the water pressure go before the pump would stop? 3. Please give me a receipt for a waterproof cement. A. The apparatus used for hydraulic elevators is similar in principle to the familiar tackle-blocks, shown in Figs. 1 and 2. In Fig. 1, where the upper pulley has fixed bearings and the lower pulley is movable and carries the weight W, it is clear that the point P of the rope will move twice as far as the weight W. In Fig. 2, there are four sections of the rope passing off from the lower pulleys, and each section must be taken up the same amount that the weight is raised; hence P moves four times as far as W. If there were three lower pulleys, P would move six times as far as W; if four lower pulleys, eight times as far, etc. It is clear, therefore, that the rope, in passing off from each additional pulley, moves further than it did in passing off from the preceding pulley, and that each successive pulley must rotate further than the preceding one. Thus, in Fig. 2, when the weight has been raised one foot, the rope at A will have

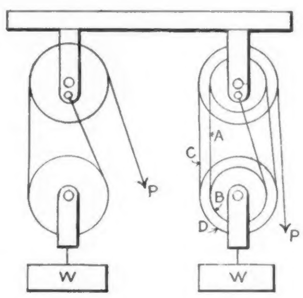


FIG. 1. FIG. 2.

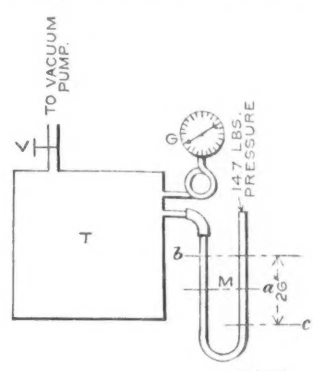


FIG. 3.

moved two feet, and the circumference of pulley B will have moved two feet, one foot being due to the rotation of the pulley and one foot due to the distance the pulley is raised. Again, the rope at C will have moved four feet and pulley D will have had a rotary motion of $4 - 1 = 3$ feet, at the circumference. This explains why the different pulleys move different distances. 2. Assuming that you mean a direct-acting steam-pump, which carries steam full stroke, the water pressure multiplied by the area of the water piston will equal the steam pressure multiplied by the area of the steam piston, when the pressure is up to the full capacity of the pump, neglecting any loss from friction. If you will examine your pump you will find that the steam piston is larger than the water piston. If there be 80 pounds pressure on each square inch of the steam piston, therefore, it is clear that the water piston, which is smaller, will be able to sustain a pressure per square inch of more than 80 pounds. 3. A cement that is much used for aquariums is as follows: Mix together one pound of litharge, one pound of clean, white sand, one pound of plaster of paris and six ounces of resin. Use sufficient boiled linseed oil to make it of the right consistency.

84. H. T. M. has leather cup packing to make in quantity. He says that after forcing into the mold the leather is trimmed and chamfered by hand, which is a slow process, and asks if there is not a quicker way to do it. A. We have not seen packing trimmed in any way other than by hand. There is no reason, however, why, with the exercise of a little ingenuity, a simple machine could not be made to do the work. Perhaps some reader has a suggestion to offer.

85. Student writes: Please answer the following questions about vacuum: How is vacuum measured? Should the measurement be in feet or inches, and in square or cubic measure? 2. What is the highest obtainable vacuum? A. Vacuum may be measured, either in pressure per square foot or pressure per square inch, but the usual way is to measure it by stating the height in inches of a column of mercury that the pressure of the atmosphere will support against the pressure of the vacuum, or more properly, the partial vacuum. Suppose we have a closed tank, T, in Fig. 3, to which is attached a vacuum-gauge, G, and a bent tube, containing mercury, at M, such an instrument being called a "manometer." At the top of the tank is a connection to a vacuum-pump. The outer end of the manometer tube is open to the atmosphere, and we will assume the valve V to be open so that the air will have free access to the tank. Under these normal conditions the mercury will, of course, rise to the same height in each arm of the manometer, as indicated by the dotted

line a. Now suppose that the connection to the air-pump be made, and that air be pumped from the tank. The pressure in the tank will drop, and the pressure of the outside air, which under ordinary conditions at the sea level is about 14.7 pounds per square inch, will cause the mercury to rise in the left-hand tube and to fall in the right-hand one. We will suppose that enough air has been pumped out so that the difference between the two levels of the mercury at b and c is 26 inches. The action will be the same, whatever the diameter of the manometer tube, but for convenience, we will assume that its inside area is one square inch. We, therefore, have an unbalanced column of mercury of one square inch cross section and 26 inches long, or 26 cubic inches in all. One cubic inch of mercury weighs .49 pounds, and 26 cubic inches $26 \times .49$, or 12.74 pounds. That is, the pressure in the tank is such that the pressure of the atmosphere can sustain a column of mercury 26 inches high against it, and is equal to $14.7 - 12.74 = 1.96$ pounds. The manometer is used for accurate determinations, and vacuum gauges are made to read in inches, like the manometer. Thus, in the present instance, the hand of the gauge would point to the number 26, and we should say that there were 26 inches of vacuum. 2. With the best air-pumps made for experimental purposes within 1-30th of an inch of a perfect vacuum can be attained; with a mercurial air-pump, within 1-250th of an inch of a perfect vacuum; and it has been estimated that a vacuum has been attained by chemical means within 1-667,000th of an inch of perfection. In steam plants, the highest attainable results are about 28 inches; 26 to 27 inches is a good vacuum.

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85. R. S. asks: Will you kindly give the proper size of standard pipe to return eight cubic feet of water per hour from a heating system to a tank trap? The pipe is horizontal and 100 feet long, but can have a drop of five inches. Pressure of steam, 80 pounds per square inch. A. We submitted this question to Mr. W. E. Hopton, mechanical engineer for Colgate & Co., Jersey City, N. J., who replied: "I take the question to be the size of pipe from a heating system, 100 feet away from the tank trap, which will have to take care of eight cubic feet of water per hour. I would use one inch pipe and connect to a trap, which would have a three-quarter-inch connection. In long distance transmission of water to or from a pump, trap, or the like, I use the next size larger pipe, unless the distance is very long."

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87. F. A. N. writes: How should a pattern be made for casting a worm? 2. Please tell me how to get the angle of the teeth for the worm-gear. A. Worms are not generally cast, but are usually turned from a solid steel bar in the lathe, which makes a much superior article. If you insist upon a cast worm, however, make the pattern in halves, with the division coming longitudinally along the axis. The only point to look out for is that the angle of the sides of the thread be great enough, so that the pattern will draw out of the sand in the mold. This point is most clearly illustrated by means of a square thread, as shown in Fig. 4. When looking at the paper in a perpendicular direction, the sides of the thread, a, a, a, cut under the top of the thread, and it is evident that a half-pattern of a square-threaded worm would not pull out of the sand for this reason. If the sides have more of an angle, however, there will not be this trouble. If you will look at a lathe lead screw and consider what shape thread is necessary to enable the half-nuts in the carriage to be withdrawn, you will understand the principle that we have tried to illustrate, as the same feature is there met with. 2. We assume that you wish to use a spur gear to mesh with the worm, but with the teeth at an angle with the face of the gear. To obtain the correct angle, lay off a right-angle triangle like that in Fig. 5, with the side A equal to the pitch of the worm, and the side B equal to the pitch circumference of the worm. The angle C is the angle that the teeth should make with the face of the gear. See the data sheet issued with the September, 1898, number of MACHINERY for further information upon worm gearing.

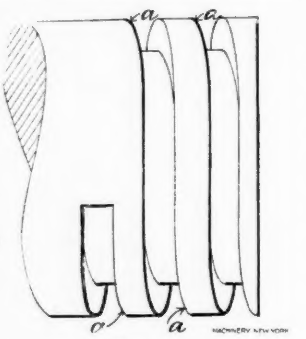

FIG. 4.

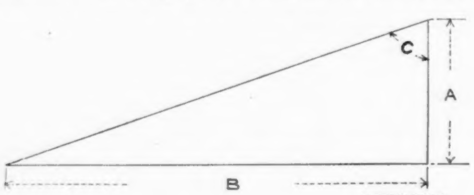

FIG. 5.

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88. H. I. encloses a piece of the hard, reddish fibre that is frequently used on drill presses, and in other places for thrust-bearing purposes, and says: "Enclosed is a sample of the material that is used in our shop for friction bearings. I would like to know what the material is of which it is composed." A. What you send we take to be a piece of vulcanized fibre, made by the Vulcanized Fibre Co., Wilmington, Del. The catalog of this company contains the following information, which, however, is not very complete: "The material is produced by treating specially prepared vegetable fibre with powerful chemical agents, whereby the exterior portion of each separate fibre becomes glutinous, and while in this condition the whole mass is consolidated and becomes practically homogeneous. After this the chemicals are extracted, the mass is manipulated, rolled, pressed and cured by various methods, and the result is vulcanized fibre. Being an extremely delicate chemical process, liable to vary with the different conditions of atmospheric moisture and temperature, it requires the utmost skill, care and experince to produce uniformly good results. The machinery required for its manufacture is cumbersome and costly, and requires to be kept in perfect adjustment at all times."

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89. C. J. encloses a plan view of a milling-machine table, on which is placed a spiral head for supporting a bevel-gear blank, on which teeth are to be cut. He says that he wants to get the thickness of the teeth and width of the spaces the same on the pitch line, and asks whether the head should be set over on the table, as shown at the left of the sketch in Fig. 6, or whether the table itself should be set at an angle, as indicated by the dotted lines. A. Neither method is correct. The bevel gear should be set with its axis running parallel with the ways of the table, and at right angles to the cutter arbor, as indicated at a in the sketch. The spiral head should be tilted in a vertical plane, of course, to bring the line of the cutting angle parallel with the table, and the depth of the spaces correct. All the lines of each tooth of a bevel gear should, if prolonged, meet at a common point in the axis of the gear. The sides of each tooth, therefore, radiate from this common point, and can be brought successively in line (horizontally) with the axis of the gear, and at right angles to the cutter arbor, as looked at from above, by simply rotating the gear. If the working side of the cutter intersect the axis of the gear, as indicated at b in the sketch, each tooth can be given the proper taper in turn without setting over, either the blank or the table.

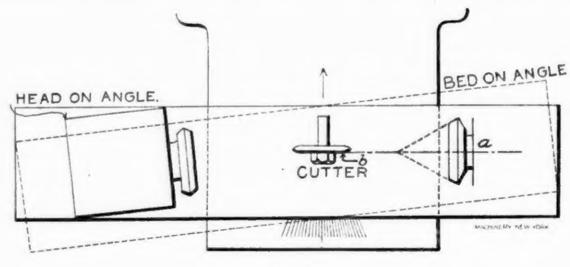

FIG. 6.

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90. F. M. asks several questions about lining up steam engines, which are not fully covered by the answer to question 65, in the last issue. We, therefore, give the following supplementary directions: A. In lining up a steam engine there are two main objects to be attained. First, the cylinder must be brought in line with and central with the guides, so that the cross-head, piston and rod will move back and forth as one body, without bending or changing their relative positions; secondly, the shaft and crank must be brought square with the center-line of the engine in all positions. We will assume that we have an engine

with bored guides and that the back head, piston and rod, cross-head and connecting-rod have been removed. The first step is to level the bed-plate by means of a level placed on the guides. Then stretch a fine cord or wire through the center of the cylinder, stuffing-box and guides, and fasten it at a point beyond the shaft. Fig. 7 shows the method of fastening the cord to the cylinder. The cross-bar is bolted to two of the cylinder studs and has a hole at the center, through which the cord passes. The cord is wound around a small stick, which is too large to pass through the hole, and which is held in place simply by the friction caused by the tension of the cord. The other end of the cord is fastened in a similar manner, either to the engine-room wall or to an upright nailed to the floor. This done, station an assistant at each end of the line, whose duties are to move the ends one way or the other, as directed, by lightly tapping the small sticks holding the line, and you are ready to proceed. The only implements that are needed for the work are a light pine stick, with a pin driven in one end, as shown in Fig.

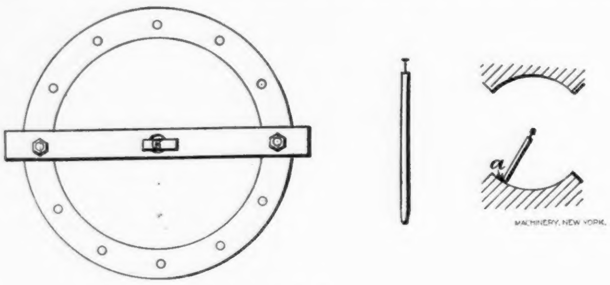

FIG. 7.

7, a few pieces of tissue paper, and a steady hand. Make a small chalk mark near each corner of both the upper and lower guide and have the stick a little longer than half the diameter of the bore of the guides. Now, by placing the stick, first on one chalk mark and then on another, try for the line, as indicated at the right in Fig. 7, and keep at work until your assistants have the line exactly central with the bore. You can bring the stick to the exact length required by tapping the head of the pin, and in case you get it too short, or the bore varies in diameter, you can make allowance by placing as many thicknesses of paper under the stick as necessary. Having the cord true with the guides, make a new stick and try the cylinder, both at the back and front ends. Do not test the front end by the stuffing-box, as this will probably not be exactly central with the cylinder. If the cylinder is found to be "out" at the front end, it should be moved bodily one way or the other. If it is "out" at the back, it should be swung around until it comes in line. If the cylinder rests on the foundation, it can sometimes be sprung the requisite amount; otherwise, shims will have to be inserted between the flanges of the bed-plate and cylinder. In case the engine is being erected in the shop, however, the flanges should be refaced, since shims are an evidence of poor workmanship. If the cylinder is "out of round," through wear or the sagging of the upper side, due allowance should be made. Finally, having brought the guides and cylinder into line, proceed to test the crank and shaft as directed in the last number of the paper. In case the engine does not have bored guides, a form of stick or "feeler" must be used, which will fit the guides, so that it will always assume relatively the same position at whatever point the line may be tried. It is very easy to tell one how to line up an engine, but it is not an easy thing to do it. It requires a delicacy of touch and an appreciation of what is meant by close measurement, that can come only through experience. In centering the line, one should be able to detect when it is as little as 1-1,000th of an inch out of center. A piece of ordinary tissue paper is about .00125 inch thick. A man should be able, therefore, to adjust a line so accurately that, if the "feeler," with one or more pieces of the paper under it, just clips the line it will miss the line when one thickness is removed. While it may not always be necessary to work as closely as this, a person cannot expect to line up engines successfully until he has a full knowledge of what this degree of accuracy means.

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91. H. J. W. writes: I have a bipolar, incandescent generator; the armature is of the drum type, 4 inches long and 3 inches diameter, wound with No. 24 double cotton-covered wire, 24 sections. The field magnet is wound with No. 26 magnet wire, and is 3 inches diameter and 3 inches long. When I run it up to 4,000 revolutions per minute, I can get 52 volts. I can only run at 3,000 revolutions per minute with the rig I have. If I put larger wire on the field magnet, will that increase my voltage by running 3,000 revolutions per minute? The armature is wound with two convolutions. Is one convolution more satisfactory in so small a dynamo? A. It is not probable that you could increase the voltage as much as you desire by increasing the size of the field wire. You would obtain some increase, because more current would pass around the field, but unless the iron is magnetized to a very low density, it will require a very great increase in the magnetizing current to effect as great a change in voltage as you desire. Small machines are generally so proportioned that the field has to be magnetized about as strongly as it pays to magnetize it, in order that they may give the necessary voltage at the speed they are intended to run. It is doubtful, therefore, whether you could gain more than 8 or 10 per cent., or possibly 15, by reducing the size of field wire. The best way to attain the object is by increasing the number of turns of wire on the armature. As the machine is small, and you perhaps do not require close regulation, it can be rewound with the same size wire, providing there is room enough at the ends of the armature core to accommodate the additional wire. If there is not sufficient room, a smaller wire must be used. You can judge by the way the wire piles up at the ends whether more of the same wire can be put on or not. So far as the periphery of the armature is concerned, it can be put on by winding three layers, instead of two, and if this makes the clearance between armature and pole faces too small, the latter can be bored out a trifle larger. The wire should be increased in the proportion of 3 to 4, or a little more. If you are compelled to use a smaller wire, the capacity of the machine will be decreased, but this you cannot help. Two layers of wire will give a greater voltage than one, and three will give more than two, as the voltage is proportional to the number of wires on the armature, and these increase with the number of turns. In large machines a single layer is often used, as it gives all the voltage required, owing to the fact that, as the field is stronger, a smaller number of turns will produce the same effect. The voltage is dependent upon the speed of the armature, the strength of the field and the number of wires upon the armature surface. If the speed be increased, the field strength can be reduced, or the number of wires can be reduced. If the field be increased, the number of wires can be reduced or the speed can be lowered, or a reduction in both can be made. In large machines the field is very much increased, therefore the speed and the wires on the armature can both be considerably reduced. The current carrying capacity is dependent upon the size of the wire.

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FRESH FROM THE PRESS.

A DICTIONARY OF ELECTRICAL WORDS, TERMS AND PHRASES, by Edwin J. Houston, A. M., Ph. D. Contains nearly 1,000 8vo pages and is illustrated by 582 engravings. Price, $7.50. Published by the W. J. Johnston Co., New York.

This extensive work will be appreciated by the student who is studying the science of electricity, and will be treasured as a valuable addition to his library. To the electrician it is simply invaluable, as everything is brought up to date, this feature being of great importance, as the phenomenal growth of this branch of engineering has brought into use so many new words and terms that books published only a few years ago are now nearly obsolete. While this work is not a text-book, there is probably no book published that will enable the busy man to get a knowledge of the principles and apparatus used in electrical appliances as quickly as will this, and besides, he is assured that the information is given by one who is considered a standard authority on these subjects.

The extent and scope of the work is such that commencing with the definition of electric, nearly twenty-six pages are devoted to kindred words, ending with the word electrum. Again, in the appendix we note that nearly nine pages are given to definitions beginning with magnet and ending with magnet wire. Duplex and quadruplex telegraphy are briefly but comprehensively explained, and about twelve pages are given to defining the terms and phrases used in telegraphy. We glean from its pages that argyrometry is the art of determining the weight of electrolytically deposited silver, and that electrozemia means a word proposed for capital punishment by means of electricity.

Some of the cuts used to illustrate the text are not very clear, but the greater part are well printed and quite satisfactory.